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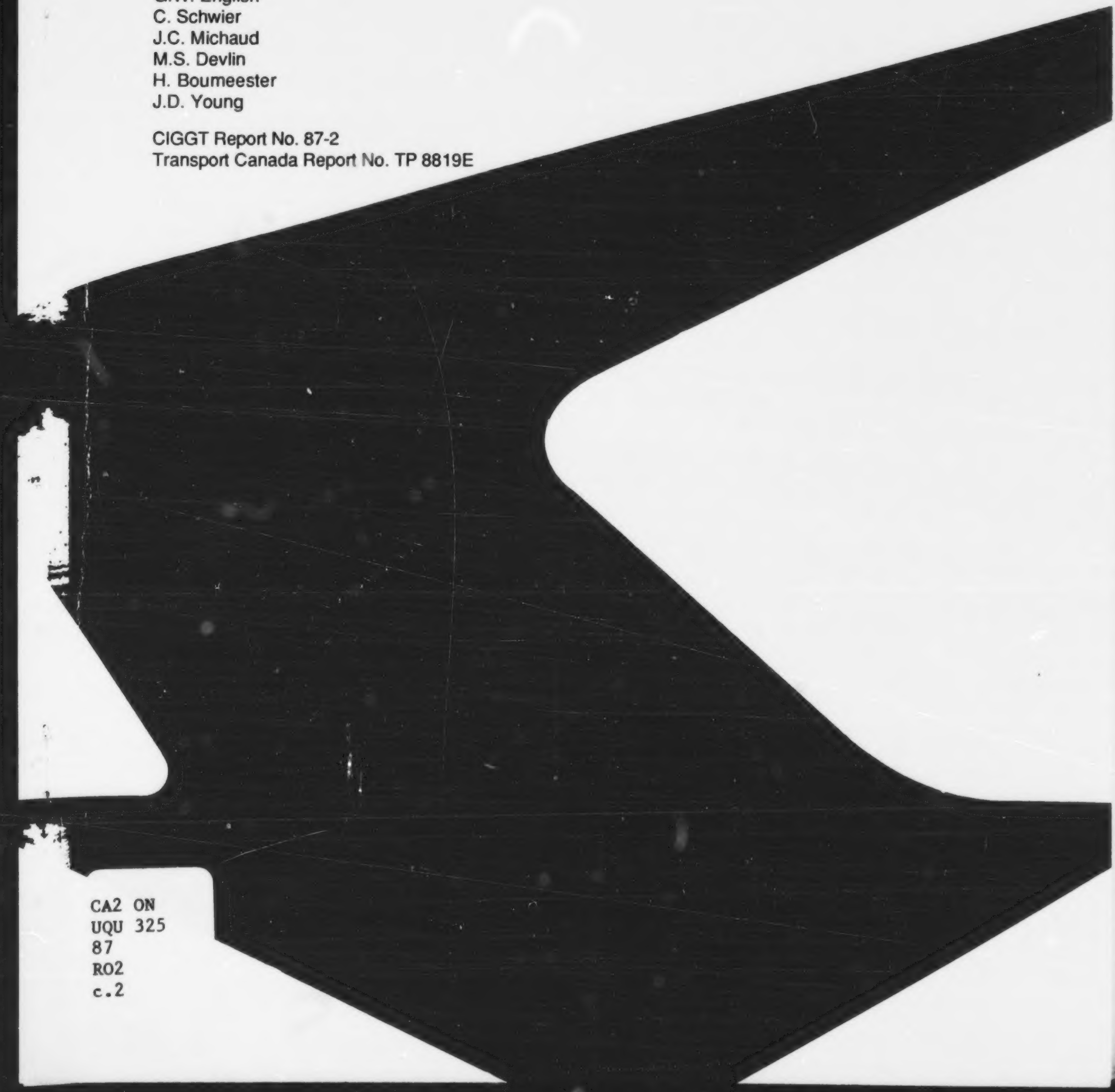
CIGGT Report No. 87-2
Transport Canada Report No. TP 8819E



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CANADIAN RAILWAY MOTIVE POWER OPTIONS AND DEVELOPMENT PRIORITIES

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Un sommaire en français de ce rapport est inclus avant la Table des matières.

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17. Abstract

Canada's two major railways face a period of major motive power renewal to the end of this century. This life-cycle cost (LCC) based investigation is an input to the decisions to be made. Analysis revealed that while current-generation locomotives demonstrate a significant incremental value over their predecessors in a buy-new comparison, they do not in a rebuild decision. In express service higher powers are necessary to make new units attractive, while in bulk service tractive effort enhancement alone is sufficient. A coordinated program of research is recommended for tractive effort enhancement. New truck development is seen as an area of opportunity for Canadian suppliers with benefits derived from the higher engine powers and tractive efforts which would be achieved. The conclusion was reached that the medium-speed diesel, although not the best in power/weight ratio, is the best choice from an LCC viewpoint to achieve high engine powers. Further investigation of the value of universality in high-performance locomotives is recommended. While ac traction permits both high speed and high tractive effort operation, the use of extra powered axles on adjacent coal cars in bulk service is recommended for dc traction. AC traction was not considered to offer sufficient advantage in a retrofit without enhanced performance but may be attractive if it can achieve increased tractive effort. Conceptual design of a steam bottoming cycle retrofit to the Bombardier four-stroke engine is recommended. No alternative fuels were considered sufficiently attractive to warrant conversion at this time. Their development is considered to be a good risk reduction investment and development of a natural gas prototype locomotive is recommended.

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17 Résumé

D'ici la fin du siècle, les deux plus importantes sociétés ferroviaires canadiennes devront avoir renouvelé ou modernisé leur parc de locomotives. La présente recherche sur le coût global des options envisageables vise à aider le processus de décision. Elle a montré que les locomotives modernes dégagent un avantage significatif par rapport aux anciennes dans un contexte où l'exploitant désire acheter du neuf, mais que cet avantage devient moins évident lorsqu'il s'agit de reconstruire. En ce qui concerne le trafic messageries, des puissances accrues sont exigées, alors que pour le trafic marchandises en vrac, un effort de traction accru peut suffire à lui seul à rendre attractif l'achat d'un matériel neuf. Un programme coordonné de recherche sur l'accroissement de l'effort de traction est recommandé. Le développement de nouveaux bogies peut devenir un créneau intéressant pour les fabricants canadiens, eu égard aux avantages qu'ils procurent : puissances et efforts de traction accrus. La comparaison montre que le moteur diesel semi-rapide, bien que désavantagé sur le plan du rapport puissance-poids, reste encore sur le plan du coût global le meilleur choix dans la recherche d'une puissance accrue. Des études plus poussées sur les avantages procurés par les locomotives universelles sont aussi recommandées. Les solutions suivantes sont suggérées : équiper d'essieux moteurs à courant continu les wagons proches des locomotives tirant des trains charbonniers ou tirer parti des caractéristiques des moteurs de traction à courant alternatif de façon à conjuguer vitesse et effort de traction élevés. Les moteurs de traction à courant alternatif posés en rééquipement n'offrent d'avantages intéressants qu'avec une puissance diesel accrue, mais ils restent une option intéressante dans la mesure où ils autorisent un surplus de traction. Il est également recommandé que soit lancée l'étude conceptuelle d'un moteur Bombardier à quatre temps partiellement converti en cycle vapeur. Aucun des combustibles de substitution envisagés n'a été jugé suffisamment attractif dans le contexte actuel, toutefois les fonds de R&D consentis à cette recherche représentent un bon investissement afin de parer à une montée rapide du diesel. Enfin, il est de plus recommandé de réaliser le prototype d'une locomotive brûlant du gaz naturel.

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EXECUTIVE SUMMARY

Canada's two major railways face a period of extensive motive power renewal to the end of this century. It is estimated that replacement or major rebuild of existing 20- to 25-year-old mainline 3,000 hp equipment will average 85 units per year. With additional traffic needs considered, the total requirement may be in the order of 100 units annually. The decisions made in this regard have significant cost implications, not only in purchase price but in lifetime operating costs. Canadian manufacturers must also decide what research and development projects involving freight locomotives, transmission systems and other locomotive components are likely to be commercially successful. For these decisions to be soundly based, it is essential that the cost-effectiveness of alternative locomotives and sub-systems be established as accurately as possible. Such is the intent of this investigation. It is a strategic evaluation aimed at producing a major motive power development program for the Canadian railways and their supply industries. The study is primarily concerned with the application of technology which is currently in use elsewhere or emerging technology which is well along the development path.

It is estimated that half of the mainline locomotive fleets belonging to Canadian National (CN) and Canadian Pacific (CP) are in adhesion dispatch service, many in the form of bulk-haul unit trains. Still, a significant proportion (25 percent) are in express service, dispatched on a power basis at about 2 hp/ton. The remaining 25 percent are also power-dispatched, but in general freight service with lower power-to-weight ratios. This balance in allocation is not expected to change appreciably in the future.

The life-cycle costs of locomotives in bulk and express service were assessed. The costs of maintenance were based on a one-year census of maintenance work performed by CP Rail on its SD40 locomotive fleet and a similar set of data from BC Rail. Fuel costs were derived from computer simulation of representative train consists over actual track profiles for CN, CP and BC Rail. These runs were used to generate "typical" duty cycles for each class of service.

PERFORMANCE ENHANCEMENT

Analysis revealed that the latest generation locomotives demonstrate a significant incremental benefit over their predecessors in a situation where the railways must choose between the two in buying new locomotives. This is the type of decision involved in meeting traffic growth demands for new motive power. However, in the much larger replacement market, the operating cost reductions to be realized with the new locomotives do not show a significant benefit over the cost of rebuilding existing locomotive units. The situation is different in the U.S. where depreciation allowances make new locomotives more attractive in a new versus rebuild comparison.

In Canada improved performance is required to show a clear edge over the rebuild option. In bulk service, tractive effort enhancement alone is sufficient to make new units attractive. This is a consequence of recent line capacity enhancements made by the railways and the very low cost of track-induced maintenance in comparison to other locomotive cost savings. A coordinated program of research into tractive effort enhancement is recommended. This includes laboratory and/or field evaluations of alumina as a friction-enhancing agent and of new wheel metallurgies. New truck development is seen as an area of opportunity for Canadian suppliers. The benefits of weight reduction and improved curve negotiation do not offer major cost savings in a retrofit to locomotives at existing performance levels. The major benefits are derived from the higher engine powers and tractive efforts which they permit. Improved weight transfer is desirable in conjunction with dc traction motors, while reduced angles of attack in curving is beneficial to asynchronous ac traction. It is also recommended that CP Rail assess the adhesion performance in curves of their prototype ac traction motor locomotive and test a simple retrofit truck interconnection as validation of the simulation-based findings of this study.

In express service, higher powers are necessary to demonstrate a clear benefit in a rebuild comparison. A number of alternative lightweight engines were evaluated and the conclusion was reached that the medium-speed diesel, although not the best in power/weight ratio, is the best choice from an LCC viewpoint to achieve high engine powers. The high-speed diesel engine realizes an estimated value in $/tractive power (tp) very close to the medium speed diesel alternatives with separate fuel tenders. However, given the extensive experience of Canadian railways with the medium-speed diesel, there would appear to be insufficient advantage to warrant a switch in technology.

A diesel-/turbine-electric hybrid was also evaluated. The turbine was chosen for its light weight, and the diesel engine was retained for its fuel efficiency advantage at low power demands. It was concluded that the turbine engine will have to demonstrate much lower maintenance costs than have been exhibited in similar duty cycles before it can be considered a candidate for railway express freight applications. Recuperators will also have to achieve much higher thermal cycle lives to be candidates in the rail application.

The development of special-purpose locomotives (i.e. bulk versus express) leads to smaller product markets from the builders' viewpoint and lower utilization from the railways' viewpoint. There is a definite value in a universal locomotive. This has been achieved with the present locomotive performance levels. However, enhanced performance in tractive effort cannot be achieved with a dc traction motor while meeting high-speed express train requirements. It is recommended that the penalties of specialization be further investigated. The operations analysis tools which would be developed in this process would also be valuable in assessing ways to increase locomotive utilization levels above the present 50 percent.

Two alternatives maintain flexibility while achieving higher performance levels. Extra powered axles can be used on adjacent coal cars in bulk service to realize additional tractive effort with dc traction. The inherent wide constant power range of ac traction permits both high speed and high tractive effort operation. Further investigation of both is recommended.

AC traction does not offer sufficient advantage in a retrofit without enhanced performance but may be attractive if it can achieve increased tractive effort. If new traction motor development is necessary to achieve the performance levels required, it is recommended that the development expenditure be focussed on ac rather than dc technology. The existing cost benefit ratio is close and cost trends continue downward for power-conditioning units. It is also recommended that maintenance experience with ac synchronous motors be obtained and that the feasibility of permanent magnet fields be investigated. The main opportunities for Canadian firms are in the area of creep control systems for ac traction to achieve maximum adhesion performance in all conditions.

Concerning the diesel engine, conceptual design of a retrofit to the Bombardier four-stroke engine with a steam bottoming cycle is recommended. Fuel savings exceeding 8 percent fuel can be realized with the conversion of two of its sixteen cylinders to a steam Rankine cycle. There are also opportunities for Canadian companies to play a part in the development of electronic fuel injection systems.

ALTERNATIVE FUELS

The benefits of alternatives to diesel fuel are tied to differential price escalation. R&D expenditures in these areas are investments in risk reduction, providing alternatives in the event of rapid diesel price escalation. They are not sufficiently attractive to warrant conversion at this time.

Natural gas was evaluated in compressed and liquefied form and as reformed methanol. Only the liquefied form is considered attractive as a conversion candidate. This is particularly true if the increased fuel volume requirement could be met on board the locomotive rather than in a separate LNG tender. CNG may offer a sufficient return in a diesel retrofit application to warrant prototype development as a means of assessing the performance of the diesel engine and to obtain operating experience with a fuel tender.

Methanol does not provide sufficient fuel cost savings to offset the added costs of operating a tender. Coal-water slurry offers similar fuel economies to that of methanol, but there are still significant maintenance cost uncertainties. These factors, combined with the existing development efforts in the U.S., give it a low priority in Canadian railway alternatives.

Another advantage of a natural gas prototype locomotive is that it addresses many of the operating concerns associated with the use of liquefied hydrogen fuel tenders. Hydrogen could become a viable alternative in the longer term. Fuel cells appear to be technically feasible in the locomotive size range and are projected to realize continued increases in power density. While the capital and maintenance costs of a railway-sized fuel cell are not known, the economics of fuel cells are largely determined by the cost of hydrogen. Hydrogen will always be more expensive than its energy source, be it electricity through electrolysis or natural gas through methane reformation. These costs are mitigated somewhat by the high fuel efficiency of the fuel cell, but are not competitive with diesel fuel at current prices. The use of off-peak nuclear generator capacity to produce low-cost hydrogen is not considered to be a situation which will be available to all hydrogen consumers on a long-term basis. Ontario Hydro has monitored hydrogen generation and recombustion as a load-leveling option. Its internal benefits of load-leveling would form the minimum market price for the hydrogen generated.

While electrification was not in the scope of this work, the value of a dual-mode electric/diesel-electric locomotive was evaluated for the mountain region. The assessment was based on a strategy of diesel propulsion for all segments requiring less than 75 percent of peak power and electric propulsion for the remaining segments. This strategy achieves much higher fuel savings per mile of electrified track than does full electrification but has other disadvantages. For bulk trains in the mountain region, the purchase value equivalent of operating cost savings per locomotive before any fixed costs are considered was below the existing and projected price of a dual-mode locomotive. It was concluded that dual-mode operation was not a viable alternative unless unusual local circumstances affect the cost savings.

SOMMAIRE

D'ici la fin du siècle, les deux plus importantes sociétés ferroviaires du Canada devront avoir renouvelé ou modernisé leur parc de locomotives; il est prévu qu'elles auront donc à décider du remplacement ou de la reconstruction des matériels de 3 000 HP ayant maintenant de 20 à 25 ans d'âge, à raison de 85 par an, voire 100 par an considérant le trafic supplémentaire prévu. Ainsi, les conséquences financières des décisions à prendre sont lourdes, non seulement pour ce qui est du prix d'achat mais aussi en ce qui concerne le coût global sur la durée de vie des équipements. Quant aux fabricants canadiens, ils devront déterminer lesquels des projets de recherche et de développement sur les locomotives marchandises, les transmissions et les autres équipements sont susceptibles de constituer, en bout de ligne, un succès commercial. C'est dire que toutes les décisions à prendre devront se faire en connaissance de cause, c'est-à-dire en fonction des avantages escomptés et des coûts correspondants de chacune des options envisagées et des systèmes qu'elles mettent en oeuvre. Tel donc a été le but de la présente recherche : évaluation stratégique ouvrant la voie à un ambitieux programme de planification des besoins en puissance de traction intéressant les sociétés ferroviaires canadiennes ainsi que leurs fournisseurs. La recherche a surtout été axée sur l'utilisation des techniques couramment utilisées ailleurs, ou encore à l'étude, mais dont le développement est nettement tracé dans la conjoncture actuelle.

On estime que : la moitié des locomotives de grande traction appartenant au Canadien National (CN) et au Canadien Pacifique (CP) sont maintenant affectées en fonction de l'effort de traction à fournir, souvent à la traction de trains-blocs; 25 p. 100 d'entre elles sont utilisées pour le trafic messageries et affectées en fonction de la puissance (2 HP/t); enfin, le restant, soit 25 p. 100, est aussi affecté en fonction de la puissance, mais généralement au trafic marchandises générales, caractérisé par un rapport puissance-poids moins élevé. Il ne semble pas que cette distribution soit appelée à être profondément modifiée à l'avenir.

La recherche a établi le coût global sur la durée de vie des locomotives utilisées en régime messageries et en régime marchandises en vrac. A cet effet, elle s'est fondée sur le coût dégagé au cours d'un an par CP Rail pour l'entretien de ses SD40 et également sur des données analogues provenant de BC Rail. Quant à la consommation, les coûts correspondants ont été tirés de la simulation assistée par ordinateur de rames représentatives circulant sur des tronçons de voie dont le profil restitue celui des voies du CN, du CP et de BC Rail. Grâce à ces simulations, des cycles de marche types ont pu être produits à l'égard de chaque catégorie de trafic.

AMÉLIORATION DES PERFORMANCES

L'analyse montre que les locomotives les plus récentes dégagent un avantage significatif par rapport aux modèles courants dans un contexte où l'exploitant doit acheter du neuf en vue de répondre aux prévisions d'augmentation de trafic. Par contre, si l'on considère le marché plus vaste des renouvellements de parcs, l'écart sur le plan du coût d'exploitation entre un matériel roulant neuf et un matériel reconstruit n'est pas évident, du moins au Canada. Aux États-Unis, la

situation n'est pas la même qu'ici, étant donné que dans ce pays une locomotive neuve peut être amortie plus rapidement, ce qui rend cette option plus attractive.

Au Canada, il faut que l'écart de performances du neuf par rapport au reconstruit soit clairement établi. En ce qui concerne le trafic marchandises en vrac, un effort de traction accru peut suffire à lui seul à rendre attractif l'achat d'un matériel neuf; ceci est une conséquence, d'une part, des efforts récents fournis dans le but d'accroître la capacité des réseaux et, d'autre part, du coût d'entretien supplémentaire de la voie entraîné par l'acquisition d'un matériel neuf, coût qui est loin de rattraper les gains que ce dernier peut apporter. Un programme coordonné de recherche sur l'accroissement de l'effort de traction est donc recommandé. Il devra porter sur l'appréciation en laboratoire et en service de l'alumine comme agent destiné à améliorer les qualités d'adhérence et des nouveaux procédés métallurgiques relatifs à la fabrication de roues. Le développement de nouveaux bogies peut devenir un créneau intéressant pour les fabricants canadiens. À eux seuls, un allégement et une meilleure inscription dans les courbes ne constituent pas des facteurs de gain tels qu'ils justifieraient un rééquipement des locomotives actuelles. Les grands avantages que ces bogies procurent résident plutôt dans les puissances accrues qu'ils peuvent transmettre et les efforts de traction supérieurs qu'ils développent. Une meilleure distribution massique est souhaitable dans le cas des moteurs de traction électrique à courant continu, ainsi qu'une meilleure inscription dans les courbes dans le cas des moteurs alternatifs. Il est en outre recommandé que CP Rail évalue les qualités d'adhérence dans les courbes apportées par la locomotive à traction électrique alternative qu'elle a réalisée en prototype, et qu'elle fasse des essais sur l'installation en rattrapage d'une liaison mécanique simple entre les bogies dans le but de valider les résultats théoriques obtenus par simulation dans le cadre de la présente recherche.

Concernant le trafic messageries, des puissances accrues doivent ressortir nettement de la comparaison entre matériel neuf et reconstruit. La comparaison entre plusieurs moteurs diesel légers a montré que le moteur semi-rapide, bien que désavantagé sur le plan du rapport puissance-poids, reste encore sur le plan du coût global la meilleure option pour l'accroissement des puissances. De son côté, le moteur diesel rapide dégage un rapport $/puissance motrice très proche du rapport dégagé par les différents types de moteurs diesel semi-rapides équipant des locomotives avec tender séparé. Cependant, les avantages qui en découleraient ne semblent pas justifier un tel changement, si on se fie à la grande expérience des sociétés canadiennes avec le diesel semi-rapide.

Une locomotive hybride diesel-électrique avec turbine a également été évaluée. La raison du choix de la turbine réside dans sa légèreté relative, celle du diesel dans sont efficacité énergétique à niveaux de puissance bas. La conclusion a été que la turbine devra dégager des coûts d'entretien bien inférieurs qu'il n'a été donné de constater jusqu'ici, dans des conditions d'exploitation similaires, avant qu'elle ne puisse devenir une option intéressante en régime messageries. Les récupérateurs devront, eux aussi, avoir une bien meilleure résistance à la fatigue thermique à laquelle ils sont astreints.

L'utilisation de locomotives spécialisées, les unes pour la traction marchandises et les autres pour le régime messageries, a également été étudiée. L'étude a montré que cela se traduirait par un fractionnement du marché dans le cas des constructeurs et un taux d'utilisation réduit dans celui des exploitants. La locomotive universelle constitue donc un choix parfaitement justifié, d'autant que les locomotives actuelles sont déjà bien placées pour remplir ce rôle. Toutefois, il faut bien le dire, les moteurs de traction à courant continu ne peuvent fournir un effort de traction accru et, en même temps, convenir aux exigences d'un régime messageries. Il est donc recommandé qu'une étude soit lancée sur les aspects négatifs de la spécialisation. À noter que les outils d'analyse de l'exploitation mis au point aux fins de cette étude pourraient également servir à évaluer les moyens d'accroître le taux d'utilisation des locomotives qui, actuellement, est de l'ordre de 50 p. 100.

Il existe deux moyens de relever les performances tout en gardant une souplesse dans l'exploitation, soit : équiper d'essieux moteurs les wagons proches des locomotives tirant des trains charbonniers, des moteurs à courant continu fournissant l'effort de traction accru nécessaire, ou tirer parti de la grande étendue de la zone d'équipuissance des moteurs de traction à courant alternatif, permettant de conjuguer vitesse et effort de traction élevés. Une étude approfondie de ces deux moyens est donc recommandée.

Les moteurs de traction à courant alternatif posés en rééquipement n'offrent pas d'avantages intéressants sans une puissance diesel accrue, mais restent une option intéressants dans la mesure où ils autorisent un surplus de traction. Si pour atteindre les niveaux de performances voulus il faut mettre au point de nouveaux moteurs électriques, il est recommandé que les travaux soient axés sur l'option alternative plutôt que continue. Même si actuellement le rapport avantages-coûts des deux options est à peu près identique, il cessera bientôt de l'être, le coût des électroniques de puissance continuant de baisser. Il est également recommandé d'acquérir de l'expérience pratique sur les moteurs synchrones à courant alternatif et d'étudier la faisabilité des moteurs à aimants permanents. En ce qui concerne les constructeurs canadiens, le créneau le plus intéressant se situerait dans le domaine des dispositifs de contrôle du glissement, appliqués aux moteurs de traction à courant alternatif en vue d'obtenir une adhérence maximale dans toutes les conditions d'exploitation.

Enfin, pour revenir au moteur diesel, il est recommandé que soit lancée l'étude conceptuelle d'un moteur Bombardier à quatre temps, dont deux des seize cylindres seraient convertis en cycle vapeur en substituant au cycle diesel un cycle Rankine, et qui serait destiné à être posé en rééquipement. Les économies de carburant réalisables dépasseraient ainsi les 8 p. 100. Quant aux constructeurs canadiens, ils y trouveraient l'opportunité d'intervenir dans le développement de systèmes électroniques d'injection.

COMBUSTIBLES DE SUBSTITUTION

Les avantages des combustibles pouvant se substituer au diesel sont liés aux écarts prévisibles des hausses de prix. Les fonds de R&D consentis dans ce domaine visent à trouver une parade à la montée rapide du prix du diesel, la substitution n'étant pas avantageuse dans le contexte actuel.

Le gaz naturel a été pris en compte, dans sa forme comprimée et liquéfiée ainsi que sous la forme de méthanol de reformage. Seule la forme liquéfiée a été trouvée avantageuse, surtout si, malgré son "excédent volumique", il était possible de loger ce combustible dans la locomotive et non pas dans un tender. Avec le gaz naturel comprimé, les économies d'énergie réalisables justifieraient la réalisation d'un prototype de locomotive en vue d'évaluer les performances avec cette forme d'énergie et aussi d'acquérir de l'expérience pratique dans l'exploitation du tender.

Quant au méthanol, les économies d'énergie ne seraient pas suffisantes pour contrebalancer les surcoûts entraînés par le tender. Les économies découlant de l'utilisation de charbon en suspension dans l'eau seraient du même ordre qu'avec le méthanol, mais il subsiste de fortes incertitudes sur les coûts d'entretien. Ces facteurs, considérés dans le contexte des efforts de R&D déjà consentis aux États-Unis dans ce domaine, placent ce produit au dernier rang parmi les combustibles envisageables par les sociétés ferroviaires canadiennes.

Un des nombreux avantages liés à la réalisation d'un prototype de locomotive au gaz naturel est que ce dernier permettra d'élucider les questions relatives à l'emploi de l'hydrogène liquide comme combustible contenu dans un tender. À long terme, ce combustible pourrait devenir intéressant. Des piles à combustible de la taille des locomotives sont techniquement faisables et il est prévu que leur puissance massique continuera de s'améliorer. Les coûts de premier établissement et d'entretien d'une pile à combustible pour usages ferroviaires sont des facteurs inconnus. Par contre, on sait que la rentabilité est largement tributaire du coût de l'hydrogène qui, invariablement, sera plus élevé que l'énergie ayant servi à le produire (procédé d'électrolyse ou conversion catalytique du méthane dans le gaz naturel). Bien que son rendement élevé contrebalance quelque peu ces coûts, la pile à combustible ne pourra pas concurrencer le diesel aux prix actuellement pratiqués. La production d'hydrogène bon marché par la mise à profit des quantités d'électricité nucléaire produites en dehors des heures de pointe ne semble pas devenir une situation susceptible de se généraliser, même à long terme. Hydro Ontario a suivi de près le projet de fabrication et de réutilisation de l'hydrogène comme option de régulation de sa courbe de charge; ainsi, le prix minimal de l'hydrogène produit par cette société serait au moins équivalent à l'avantage financier tiré de cette technique de régulation de la courbe de charge.

L'électrification n'appartenait pas au domaine de la recherche; toutefois, l'utilité de la locomotive bimode (électrique et diesel-électrique) a été évaluée dans un contexte d'exploitation en région montagneuse. Cette évaluation a été faite sur une utilisation partagée entre la propulsion diesel, là où la puissance mise en oeuvre représentait moins de 75 p. 100 de la puissance de pointe, et la propulsion électrique pour le reste. Un tel régime d'exploitation permet des économies de combustible par mille de voie électrifiée plus fortes que l'électrification complète, mais il comporte des inconvénients. Pour la traction de marchandises en vrac en région montagneuse, les économies directes d'exploitation par locomotive, c'est-à-dire avant la prise en compte des coûts fixes, sont inférieures au prix actuel et prévisible d'une locomotive bimode. D'où la conclusion que la bimodalité n'est viable que si des circonstances extraordinaires viennent affecter le calcul des économies réalisables.

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1. INTRODUCTION

Canada's two major railways face a period of extensive motive power renewal to the end of this century. It is estimated that replacements or major rebuilds of existing 20- to 25-year-old mainline 3,000 hp equipment will average 85 units per year. With additional traffic needs considered, the total requirement may be in the order of 100 units annually. The decisions made in this regard have significant cost implications, not only in purchase price but in lifetime operating costs. Canadian manufacturers must also decide what research and development projects involving freight locomotives, transmission systems and other locomotive components are likely to be commercially successful. For these decisions to be soundly based, it is essential that the cost-effectiveness of alternative locomotives and sub-systems be established as accurately as possible. Such is the intent of this investigation. It is a strategic evaluation aimed at producing a major motive power development program for the Canadian railways and their supply industries.

There have been a number of evaluations of alternative motive power systems in recent years, most of them with a focus on engines. The Jet Propulsion Laboratory (JPL) undertook a broad review of alternative forms of engines and fuels for railway motive power in 1981,[1] drawing largely from a previous analysis of the automobile.[2] The Transportation Research Board undertook a review of several alternative engines for the Association of American Railroads.[3] Three contenders were evaluated -- gas turbines, fuel cells and electrification. The Association of American Railroads (AAR) also initiated a review and forecast of diesel-electric locomotive developments to the year 2000.[4] The U.S. Department of Energy has stimulated development activity directed towards the use of coal as a fuel in railroad locomotives.[5,6,7] Previous motive power evaluations in Canada have dealt with electrification[8] and hydrogen-fuel cells.[9]

The scope of this report includes assessing system options, technical developments, improvements in performance and reliability, first costs (in current-dollar terms), relative fuel costs (over a range of energy prices), and maintenance costs for various locomotives (and "hybrids") for freight railways to the year 2000, thus prioritizing R&D efforts for pursuit of those areas of most promise. With a time horizon of only fifteen years, the study is primarily concerned with the application of technology which is currently in use elsewhere or emerging technology which is well along the development path.

This study is concerned with the complete locomotive. Engines, fuels, transmission systems and trucks are assessed from two viewpoints. The first is the establishment of the relative value in economic terms of enhancing the performance of individual components of the existing diesel electric locomotive. The second is a review of the technological options available for realization of the performance improvements. The assessment considers Canadian railway operating conditions.

1.1 Section Overview

Section 2 discusses Canadian railway operating conditions and presents the base assumptions concerning motive power operating requirements to the year 2000.

Section 3 presents the life-cycle cost methodology used in the evaluations, and outlines the development of the base-case life-cycle cost components.

In Section 4, the value of individual component improvements (e.g. engine, traction motor) to the base-case locomotive are assessed. Section 5 assesses the value of performance enhancement, in terms of tractive effort and power, beyond that of the base-case locomotive, and investigates the implications of increased specialization of locomotive design.

Section 6 then addresses the constraints to achieving higher tractive effort performance and presents mitigative measures. In Section 7 a similar assessment of alternatives to achieving higher power locomotives is undertaken.

Section 8 considers a number of alternative fuels for locomotives. In the final section, conclusions are drawn and a number of research recommendations are made.

2. PRESENT RAIL MARKET AND MOTIVE POWER ALLOCATION

In terms of both tonnage and ton-miles, bulk commodities dominate Canadian railway operations. Several individual commodities (coal, sulphur, potash) are concentrated to a small number of origin-destination movements where unit train operations can be used. These three commodities account for 24 percent of the Canadian Class I railways' total ton-miles. It is estimated that for grains, half involve mainline movement in dedicated trains under motive power requirements similar to those of unit trains.

These movements are very efficient in terms of power and energy utilization. The net-to-gross weight ratios are high and the power-to-weight ratios of the train are low. Terminals are designed for rapid turnaround to achieve high equipment utilization.

At the other end of the spectrum is express traffic. This comprises time-sensitive, truck-competitive movements which run to a schedule. While express movements are estimated to account for 20 percent of total ton-miles, the range of car types and origin-destination pairs does not permit extensive unit train operation. Intermodal shipments (trailers and containers) account for roughly one-third of the express traffic and permit dedicated trains in two markets: trailers between Montreal, Toronto and Chicago, and containers between Toronto, Montreal and Atlantic ports. In most cases express trains are made up in classification yards from a mix of priority shipments.

From a motive power viewpoint, express traffic is much less efficient. The shipments have low net-to-gross weight ratios and train power-to-weight ratios are high. To improve utilization, locomotives are drawn from a general service pool for a specific run rather than dedicated to express only.

The remainder of the traffic may be classified as general freight. Trains are made up in classification yards and linehaul motive power may be allocated on either an adhesion or a power basis, depending on the priority of the train class. The average utilization of motive power is between that of express and bulk.

Table 2.1 estimates the allocation of 3,000 hp (or greater) mainline locomotives to these three traffic types on the basis of 1984 traffic levels. With general freight further broken down into 35 percent adhesion-dispatched and 65 percent power-dispatched, there are 670 units in adhesion service, 365 units in high power (2 hp/ton) express service, and 360 units in average power (1.2-1.5 hp/ton) freight service.

The service breakdown is largely an accounting exercise, as both railways draw from a general motive power pool rather than designating locomotives for a particular service. In CP's case, where 1 percent or greater controlling grades exist across the whole system, six-axle locomotives constitute 100 percent of the pool. CN, with 0.4 to 0.6 percent grades on much of its system, uses mainly six-axle locomotives in the West and mainly four-axle locomotives in the East.

TABLE 2.1

MAINLINE MOTIVE POWER ALLOCATION, 1984*

Traffic Class	Measurement			Annual Locomotive Productivity (10^6 per unit)	Locomotive Distribution (No. of units)
	Traffic Distribution (% of NTM)	GTM/NTM	Inferred GTM (10^9)		
Bulk	36	2.7	96	205	470
General freight	48	2.0	100§	180	555
Express	16	2.4	60	165	365
Total	100	n/a	256§	n/a	1,390

* Based on 157 billion net ton-miles.
§ For general freight, only two-thirds are on main line.

This standardization of motive power may change in the future. Over the past decade the railways have found it economical to switch from a general-purpose boxcar to many specialized car designs. The lower utilization which accompanies specialization has been offset by more economical movement of specific commodities. It is natural that locomotives with much higher capital costs and maintenance parts inventory costs would lag cars in a shift to specialization. However, the economics of motive power specialization may be favourable in the near future.

2.1 Market Developments

Until recently the growth forecasts for Canadian rail traffic have been largely focussed on western Canada and based on significant growth in thermal coal exports. The 12 percent per annum forecasts of 1981[10] have been lowered each year since. Recent Transport Canada forecasts[11] are for an average 4.2 percent growth rate, derived from a 2.2 percent estimate for metallurgical coal and 10 percent for thermal coal. The projections of a major study of world coal trade[12] for the year 2000 represent a range of growth rates from 1.6 to 5.2 percent per annum for Canadian exports.

These reduced forecasts for various coal exports still have a considerable risk element. While the growth projections of world demand have a high level of confidence, the ability of Canada to maintain its market share is less certain. Various off-shore suppliers, with much lower delivered costs to both Europe and Japan, could obtain a significant portion of Canada's expected thermal market.

China may also be in a position to displace Canadian exports to Japan. Thus, even the much reduced forecasts of today may be optimistic. The pessimistic scenario growth rate of 1.6 percent per annum may be equally probable to the 4.2 percent Transport Canada forecast.

The other principal bulk commodities (grain, sulphur and potash) have been forecast by Transport Canada to grow at 2.5, 2.5 and 4.1 percent, respectively.

Again, there is considerable risk with respect to the ability of Canada to retain its market share. In fact, the growth in traffic will in some ways depend on costs of production and shipping. The same study forecasts all other commodity rail loadings to grow at 1.1 percent per annum.

From a motive power perspective, the relative performance of the intermodal market is important. In 1983, Canadian Class I and Class II common carrier truckers transported 134 billion tons.[13] However, the major part of the truck market is inaccessible to rail. Of the total truck tonnage, 83 percent involved intra-provincial movements with an average distance of 151 miles. The largest concentration of inter-provincial traffic is in the Toronto-Montreal regions, where 6.7 million tons are transported 590 km. The remaining 12 percent of the traffic is shipped inter-provincial at an average distance of 572 miles.

The rail intermodal market, on the other hand, is 99 percent inter-provincial, with an average shipment length of 1,000 miles.

Unfortunately for rail, the largest market concentration (Toronto-Montreal) is close to the truck/rail break-even distance. Rail must charge below truckers' direct operating costs to encourage them to ship by rail rather than make some contribution to their fixed costs. Rail attracts less than 2 percent of the common carrier shipments here but their total market share appears to be about 10 percent.

Only radical, but reasonable, changes in costs or regulations would impact the market balance in this region.

In longer-distance markets (e.g. Toronto-Montreal to Winnipeg-Edmonton/ Calgary-Vancouver) rail has a definite cost advantage while achieving similar time performance. Rail attracts about 20 percent of common carriers, and in total handles over 50 percent of trailer traffic in these markets. It is difficult to rationalize the large share truckers attain in these long-distance markets. However, the total size of each point-to-point market is less than a train load a day for each railway, so even full penetration of these markets would not lead to dedicated unit train service (a la HPIT).

Both railways have made, and continue to make, substantial investments in intermodal terminals and equipment. An average growth rate in excess of 5 percent per annum over the past seven years has been achieved. While the rail intermodal market is not entirely linked to the aggregate trucking market, it is driven by the same economic factors. More than half of the railway express market is tied to domestic manufacturing activity, as is the trucking market. A growth rate of half that of trucking seems pessimistic. The Transport Canada forecast of 1.1 percent applies to intermodal, yet the forecast for the trucking sector is 2.4 percent. It is reasonable to expect the railways to at least match trucking growth in the future. More importantly, a commitment to this market indicates a desire to continue to reduce direct operating costs through improved equipment. Motive power can play an important part in this areas.

2.2 Related Technology and Attitude Influences

The main non-motive power technology factor expected to influence motive power requirements to the end of this century is the Association of American Railroad's (AAR's) High Productivity Integral Train (HPIT) concept. This program is encouraging new systems-perspective equipment designs for dedicated non-interline service. The main target is the highly competitive intermodal market where 50 percent reductions in operating costs are being sought.

From the Canadian viewpoint, more important than the HPIT itself is the attitudinal change within the railway industry which led to its adoption. As noted previously, the market size in Canada is not considered sufficient to warrant large-scale unit train service for intermodal traffic. However, the initiative of the railways in adopting HPIT objectives is very important to motive power design. It indicates a willingness to relax widely accepted Federal Railroad Administration (FRA) interchange design criteria where service-specific applications do not demand it. It also signals the recognition of full life-cycle costs as the decision criterion for equipment purchases. This is important in the Canadian situation because the size of both the bulk and express markets is sufficient to consider special-purpose locomotives. HPIT is a signal to the supply industry that an express locomotive may not be more than a bulk-haul locomotive with higher-speed gearing. If the economics warrant it, significant departures from traditional design criteria will be accepted, provided the design criteria are as stringent for the intended application as the FRA standards are for interchange applications. This is not interpreted as a departure from conservatism. It is highly unlikely that this century will see the adoption of technologies not already proven elsewhere.

The degree of specialization will be highly tied to purchase volume, and in turn to its influence factors. From the railway viewpoint a pool of 200 or more special-purpose locomotives would be required to get the economies of scale associated with the dedicated maintenance facility. From the builder's viewpoint, order sizes in the 50s would be required to achieve reasonable economies of production and design for variations of existing technology; ongoing orders of even greater size would be required to undertake the development of radically different technologies.

The growth-induced locomotive orders (30 or less per annum for both CN and CP from all suppliers) is unlikely to stimulate any major change on its own. An addition of 90 units per year from a 20- to 25-year-old renewal market could generate the necessary levels of production. The realization of this market is, at the moment, quite uncertain. The cost effectiveness of an in-house rebuild versus a new equipment purchase is in the balance. The capital and operating economies of service-specific designs may be necessary to weight the balance in favour of new equipment purchases. Yet, the provision of these designs is in turn dependent on the existence of the market.

An important factor in the final determination of this "catch-22" situation may be the deregulation of American railways. With deregulation, competition in certain transborder markets has intensified. One of the factors which has been highlighted in the ensuing comparative examination is the advantage which American railways have in taxation depreciation of rolling stock and motive power. For locomotives, American railways are allowed a five-year write-off, while Canadian railways have a 10 percent declining balance provision. This is a large factor in the "build versus buy" decision. It effectively adds 25 percent to the capital cost of the "buy" decision, and partially explains a 12-year replacement policy for many American railways versus a possible 20-year rebuild decision by Canadian railways. A move to give the Canadian railways equivalent tax write-off provisions may lead to a "buy" decision on 20-year-old locomotives. In any event, it was deemed appropriate to adopt the 20-year rebuild of existing SD40 and GP40 locomotives as the basis of comparison in the evaluation of alternatives.

2.3 Locomotive Service Cycle

The function of mainline locomotives is to provide linehaul service for designated rolling stock between yard or terminal points. Yet its utilization in this role is limited by a number of factors. A study of Class I U.S. railroads[14] found that "the majority of the carriers have a locomotive utilization rate of less than 40 percent." The remainder of the time is composed of downtime for repair or maintenance, servicing (i.e. sand and fuel) and inspection, and terminal time (composed of set-off and pick-up activities as well as idle stand-by in a service pool inventory). The relative magnitudes of these components of the motive power service cycle are shown in Figure 2.1. These numbers represent system averages. Significant variations exist by type of service and region, particularly for "stand-by" time. For example, units in a 2,000-mile linehaul service between major yards will realize a much higher ratio of linehaul/yard time than will units in a 600-mile bulk haul between load and unload terminals with no other traffic originating. As most units operate out of a general service pool, the regional averages are a reasonable reflection of the service cycle it will see.

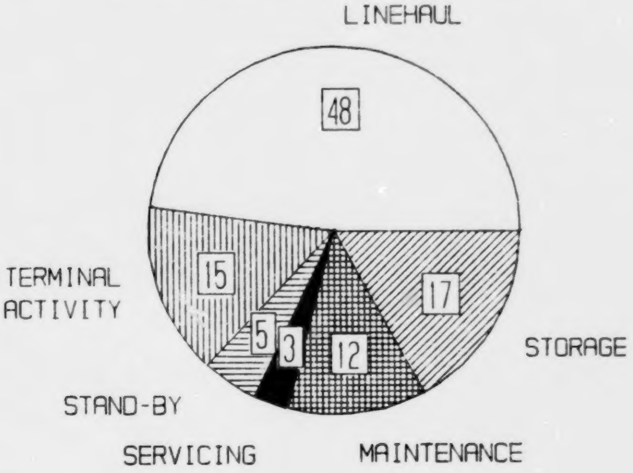

Figure 2.1 Locomotive service cycle

As the present study is concerned with the sensitivity to reductions of flexibility and utilization from service-specific applications (e.g. bulk and express) and larger unit block sizes, more detailed service cycles were derived. The estimated characteristics for each service category are presented in Table 2.2.

2.4 Locomotive Duty Cycle

Locomotive duty cycles are represented by the percentage of utilized time spent in the various throttle notch positions. "Light-," "medium-" and "heavy-duty" characteristics have been derived by General Motors' Electro-Mechanical Division (EMD) by monitoring aggregate notch times for their mainline locomotives in various railway services. The "heavy-duty" cycle has been used as a basis of comparison in the Transportation Research Board (TRB) investigation of alternative sources of motive power,[15] and in other studies. It has therefore been included in this study, but is not taken as the base reference. As one of the main objectives of this investigation was to evaluate service-specific alternatives, duty cycles were derived for each service of interest. This was accomplished via computer simulation. Representative trains for each service class were run on the mainline routes of both CN and CP. The characteristics of these trains are summarized in Table 2.3. Notch times were then aggregated for each train run and duty cycles were derived for a round trip in each case. As the relative use of automatic air brakes and dynamic brakes cannot be derived from simulation, they were assumed to be used in 25 percent of the brake applications for loaded bulk trains and in 10 percent of express train brake applications.

TABLE 2.2

LOCOMOTIVE SERVICE CYCLES

	Express	General Through Freight	General Local Freight	Bulk Unit
Average trip length (miles)	1,000	800	800	800
Average schedule speed (mph)	30	25	18	20
Terminal time at start up (hrs)	4	4	4	4
Terminal release time (hrs)	5.5	5.5	5.5	5.5
Total cycle time (hrs)	42.8	41.5	53.9	49.5
Annual miles	130,000	100,000	80,000	115,000
In-service utilization (%)	63	59	61	81
Road time utilization (%)	48.6	45.5	50	65
Maintenance down time (%)	12	12	12	12
Time in service pool pool inventory (hrs/trip)	5.5	7	8	N/A
Implied seasonal shut down (%)	16	19	17	7

Table 2.4 compares the derived duty cycles for a bulk unit train in the mountain region to those of an express train in a Toronto-Vancouver run whRich includes mountain, prairie and Canadian shield regions. The bulk train, which has a very large difference in loaded direction to empty direction trailing weights, demonstrates a large variation in the respective duty cycles. The express train, on the other hand, has a much smaller variation in trailing weights by direction and is influenced more by aerodynamic drag than the bulk train. As a result, the total round-trip duty cycles are very similar, even though the services are quite different. These figures reflect the "in motion" notch-time distribution only.

This investigation is concerned with tractive effort performance as well as the engine and fuel aspects of motive power. Thus, the tractive effort duty cycle was also derived. Figures 2.2 and 2.3 illustrate the tractive effort distribution associated with CN's and CP's bulk movements in the mountain region.

CP's bulk movement through the mountains places the most severe demands on tractive effort. In all other situations, the peak tractive effort demands are smaller in magnitude and of a much shorter duration. The coincidence of track curvature and maximum tractive effort demands was also checked. In mountain operations, CP must develop 95 percent of its peak tractive effort requirements while negotiating curves of up to 8° (218 metres). On CN's route the worst case is a 4° curve.

In order to get the total service distribution, idle time must be incorporated. Total idle time was derived on the basis of locomotive utilization figures of 100,000 miles/year for bulk service and 130,000 miles/year for express service. The idle component was then further categorized as 20 percent in-service idle and 80 percent extended time or layover idle. The study-based duty cycles are compared with the EMD "heavy-duty" cycle in Table 2.5.

TABLE 2.3

CHARACTERISTICS OF REPRESENTATIVE TRAINS

Region	Direction	Train	Locomotives	Cars
CN Mountain	West	Bulk	3 SD40	98 @ 130 tons
	East	Bulk	3 SD40	98 @ 30 tons
	West	Express	3 SD40	68 @ 56 tons, 7 @ 25 tons
	East	Express	3 SD40	68 @ 56 tons, 7 @ 25 tons
	West	General	3 SD40	96 @ 80 tons, 11 @ 25 tons
	East	General	3 SD40	78 @ 80 tons, 29 @ 25 tons
CP Mountain	West	Bulk	5 SD40	108 @ 130 tons
	East	Bulk	5 SD40	108 @ 30 tons
	West	Express	4 SD40	64 @ 56 tons, 7 @ 25 tons
	East	Express	4 SD40	64 @ 56 tons, 7 @ 25 tons
	West	General	4 SD40	85 @ 80 tons, 9 @ 25 tons
	East	General	4 SD40	47 @ 80 tons, 31 @ 25 tons
CN Prairie	West	Grain	3 GP40	106 @ 120 tons
	East	Grain	3 GP40	106 @ 30 tons
	West	Express	3 GP40	68 @ 56 tons, 7 @ 25 tons
	East	Express	3 GP40	68 @ 56 tons, 7 @ 25 tons
	West	General	3 GP40	96 @ 80 tons, 11 @ 25 tons
	East	General	3 GP40	78 @ 80 tons, 29 @ 25 tons
CP Prairie	West	Grain	4 SD40	108 @ 130 tons
	East	Grain	4 SD40	108 @ 30 tons
	West	Express	3 SD40	64 @ 56 tons, 7 @ 25 tons
	East	Express	3 SD40	64 @ 56 tons, 7 @ 25 tons
	West	General	4 SD40	85 @ 80 tons, 9 @ 25 tons
	East	General	4 SD40	47 @ 80 tons, 31 @ 25 tons

TABLE 2.4

IN-MOTION DUTY CYCLE
(percent)

	Bulk/Mountain			Express/Cross-country		
Idle	Loaded	Empty	Total	Main	Return	Total
Idle	21	11	17	12	12	12
1	1	2	1	2	3	2
2	1	5	3	4	5	4
3	2	8	5	6	8	7
4	2	12	7	7	8	8
5	3	10	6	7	8	8
6	3	11	7	6	8	7
7	3	10	6	6	7	6
8	64	31	50	50	41	45

TABLE 2.5

TOTAL SERVICE DUTY CYCLE
(percent)

	Bulk/Mountain	Express/Cross-Country	EMD Heavy-Duty
Dynamic	2	1	9
Idle	9	8	8
1	1	2	3
2	2	3	3
3	3	5	3
4	5	5	3
5	4	5	3
6	5	5	3
7	4	4	3
8	34	30	30
In-Service Idle	32	32	32

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* adjusted on the basis that 80 percent of the idle time in the EMD duty cycle is non-linehaul

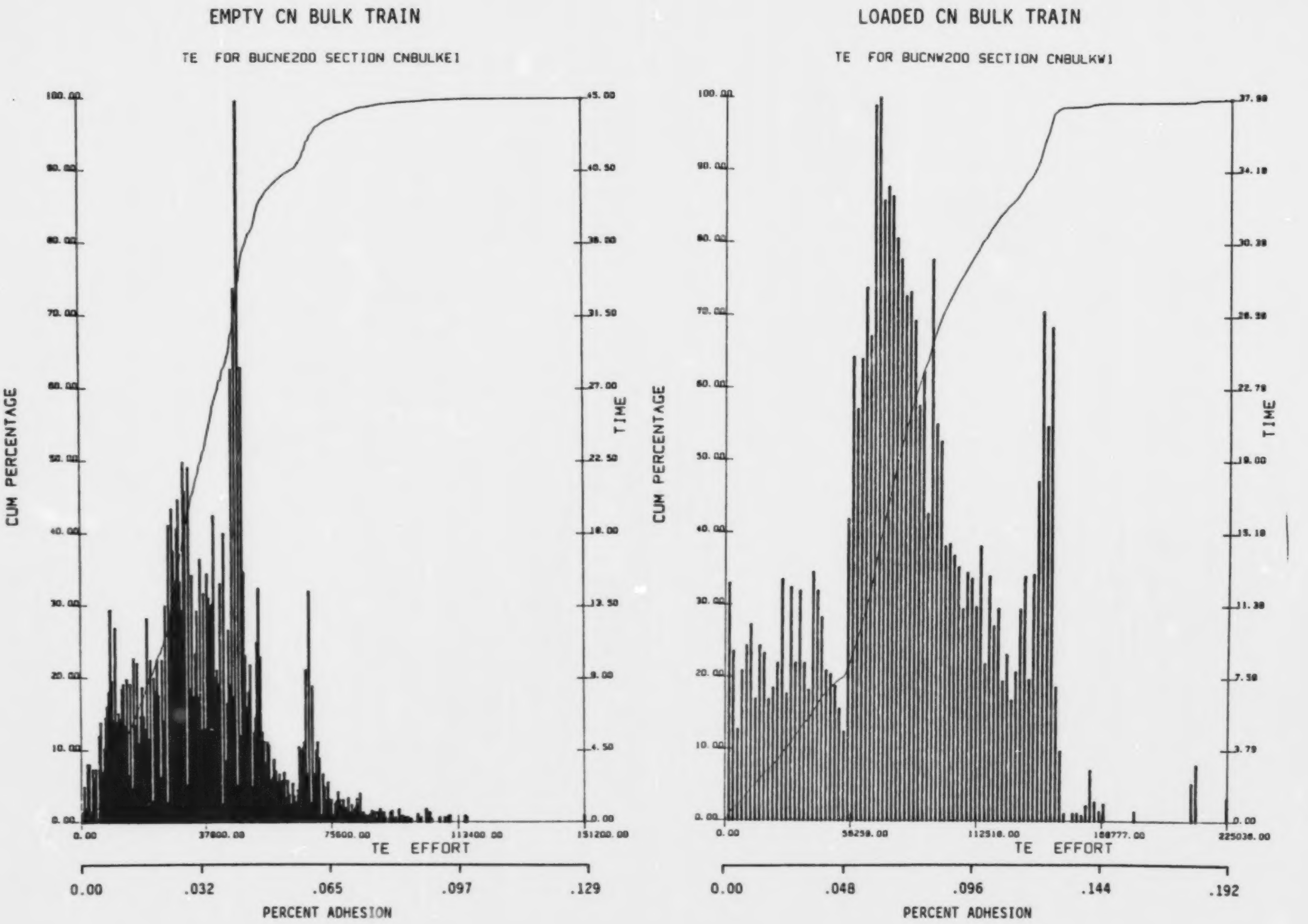


Figure 2.2 Tractive effort duty cycles, CN Mountain Region

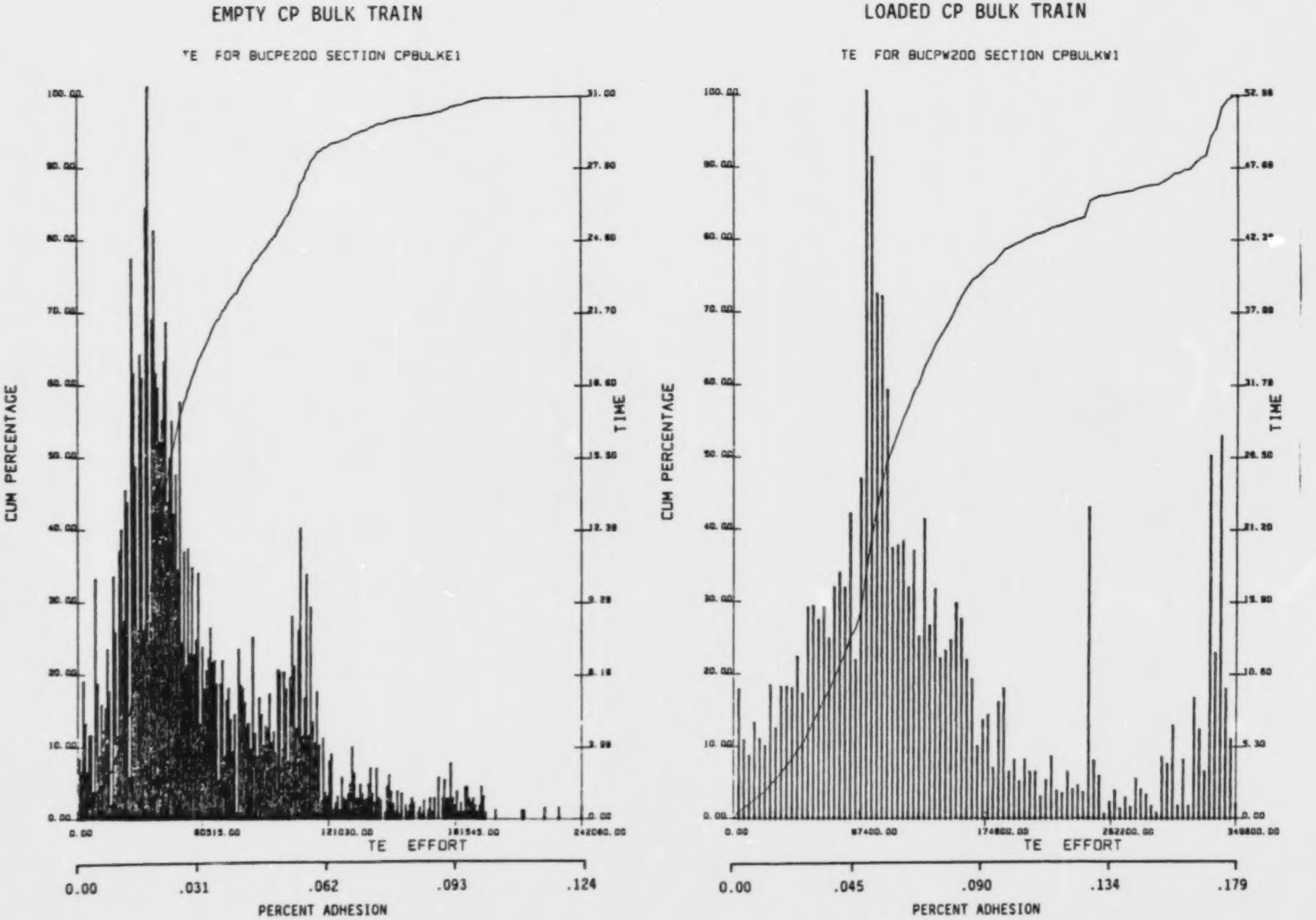


Figure 2.3 Tractive effort duty cycles, CP Mountain Region

3. ECONOMIC EVALUATION METHODOLOGY AND BASE-CASE LIFE CYCLE COST

3.1 Life-Cycle Cost Basis

The financial evaluation procedure used to determine the attractiveness of motive power developments is based on a life-cycle costing of direct motive-power-related components of the railway system. Among such costs are: locomotive acquisition cost, periodic overhauls, routine maintenance (both scheduled and unscheduled), servicing, road failures, fuel and track maintenance. Annual cashflows for these categories are projected over the 20-year expected economic life of a locomotive, given the requirements of the various example services described in Section 2. Projected changes in the real -- net of general inflationary pressures -- level of input prices over this period are incorporated into the calculations. Annual cashflows are then discounted to determine the Present Value of the direct motive power costs for the service in question. Also incorporated into the cashflow discounting process are taxation considerations, including such elements as Capital Cost Allowance savings. Depreciation, being an accounting rather than a cash flow, and financing options are not considered.

In addition to the direct costs, there are various costs which are indirectly related to motive power developments such as crew, those related to train-miles and caboose-miles, and car fleet size. Since alternate motive power developments can result in changes in train speeds, consists and in the number of trains required to move a given volume of traffic, estimates of these costs have also been included.

One potentially major indirect cost related to some of the motive power developments considered has not been directly included in the quantification. This is the effect on line capacity and interference costs to other trains resulting from changes in train characteristics and the number of trains. These factors were evaluated in a separate analysis.

The 3,000 hp, six-axle SD40-2 locomotive is used as the base case for both the bulk service option and the express service option. The equivalent four-axle locomotive has also been examined in the express service option. Two variants of the base case are considered -- purchase of a new unit and rebuilding of existing units to give equivalent to new performance.

For each motive power alternative, the costing/discounting process is repeated, taking into account differences in maintenance and fuel requirements and any changes in train composition, the number of trains required, as so on. Using the present value of locomotive-related costs for the base case and the alternate, the price per new locomotive unit that would leave the railway indifferent between continued use of the base case or adoption of the alternate is then determined. This is the capital value that yields the same present value of costs for the alternate as the base case. The procedure recognizes differences in locomotive fleet size due to differences in the haulage capacity and to changes in average unit availability stemming from different maintenance requirements.

The major components of this economic modeling procedure are illustrated in Figure 3.1.

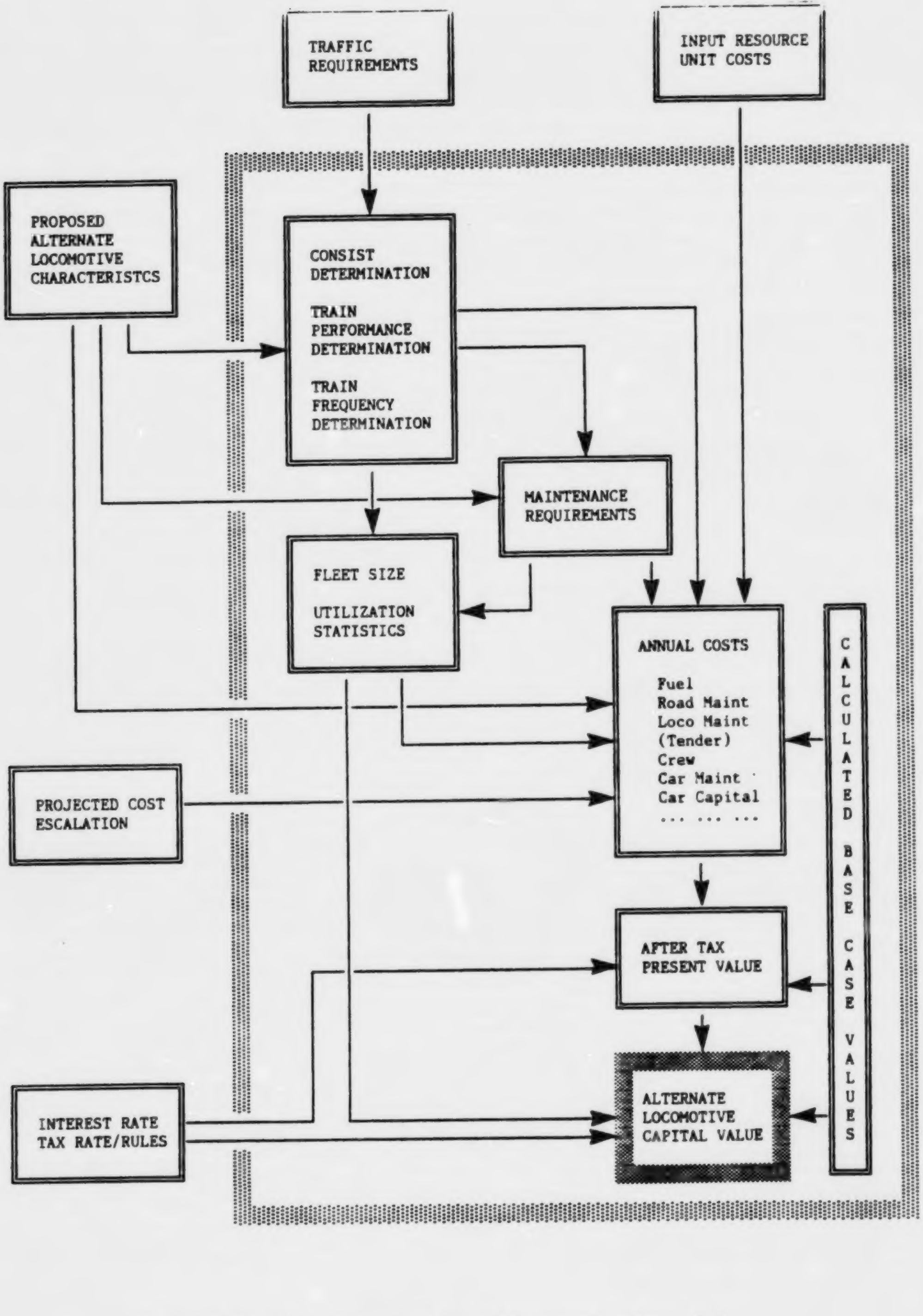

TRAFFIC REQUIREMENTS
INPUT RESOURCE UNIT COSTS
PROPOSED ALTERNATE LOCOMOTIVE CHARACTERISTCS
CONSIST DETERMINATION
TRAIN PERFORMANCE DETERMINATION
TRAIN FREQUENCY DETERMINATION
MAINTENANCE REQUIREMENTS
FLEET SIZE
UTILIZATION STATISTICS
ANNUAL COSTS
Fuel
Road Maint
Loco Maint
(Tender)
Crew
Car Maint
Car Capital
... ...
CALCULATED BASE CASE VALUES
PROJECTED COST ESCALATION
AFTER TAX PRESENT VALUE
INTEREST RATE TAX RATE/RULES
ALTERNATE LOCOMOTIVE CAPITAL VALUE

Figure 3.1 Major components of the economic model

3.2 Risk Evaluation

While this study is concerned with technological alternatives which are currently available or in development, there are still varying degrees of uncertainty in relation to their application to the Canadian railway operating environment. This is a "technological risk" factor. Another element of risk which is considered in the evaluation is the "business risk," or degree of uncertainty associated with the future operating environment. There are two categories. The first is the operational requirement for motive power over its useful life. There is considerable uncertainty surrounding the traffic forecasts. A special-purpose bulk-haul or express-service locomotive will not realize the same benefits in a different traffic allocation. Thus, if bulk export or express traffic declines, the return on investment associated with the locomotive will also decline. The second category is the unit-price expectations for the various operating costs. Fuel prices are the most notable factor in this category. These risks have been incorporated into the analysis by performing sensitivity analyses on parameters which contain a large degree of uncertainty.

3.3 Base-Case Life Cycle Cost

The North American railway industry has adopted the diesel-electric locomotive as the dominant form of motive power. In Canada, the mainstay of linehaul operations is the 3,000 hp locomotive, primarily the six-axle SD40-2 and the four axle GP40. With over 15 years operating experience, the physical and economic performance of these locomotives are well documented. While there is a newer generation of locomotive available with known physical performance, there is insufficient maintenance and reliability experience to treat these units as a base case. This section presents the economic performance of the study base-case locomotive.

3.3.1 Life Cycle Cost Components

The costs incurred in the operation of a locomotive throughout its service life include its purchase price, maintenance and servicing costs, service disruption costs of in-service failure, fuel consumption costs and induced track maintenance costs. These costs are described in this section for the base-case locomotive. As many of the alternatives to be evaluated involve component changes only, the costs are broken down by component where applicable. All costs are reduced to equivalent net present value (NPV) for aggregation and comparison. The base-case assumptions for the NPV calculations are shown in Table 3.1.

3.3.2 Summary Life-Cycle Costs

The development of each life-cycle cost component is discussed in detail in the following sections. The resulting values are summarized in Table 3.2 in relative percentage terms. It is evident that the major cost item (nearly half) is fuel. The cost of maintaining and servicing a locomotive over its life time is close to the cost of original purchase (roughly 23 percent of total). The smallest component of its life cycle cost is its induced track maintenance costs at 2.7 percent.

TABLE 3.1

KEY ECONOMIC ASSUMPTIONS FOR LIFE-CYCLE COST ANALYSIS

Average Real Cost of Capital Rate	8.0 percent (exluding inflation)
Average Long-Term Inflation Rate	4.0 percent
Real Labour Price Escalation (includes productivity impact)	A 1.5 percent decrease to 1990 followed by modest gains in the mid-1990s
Real Diesel Fuel Price Escalation	Increasing slowly to 1.0 percent annual escalation in the late 1990s.
Parts Price Escalation	Not significantly different from general inflation
Average Corporate Income Tax Rate	52 percent
Capital Cost Allowance for Locomotives	10 percent

TABLE 3.2

DISTRIBUTION OF LOCOMOTIVE-RELATED LIFE CYCLE COSTS SIX-AXLE LOCOMOTIVE IN EXPRESS SERVICE

(20-year life, 130,000 miles utilization/year)

	Capital	Overhaul	Annual Maintenance	Servicing /Inspection	Road Failures	Total
Engine	7.5	2.2	2.1	0.1	0.2	12.1
Traction motors	4.6	0.2	3.9	0.0	0.2	8.9
Electrical system	3.9	0.6	0.2	0.0	0.3	5.0
Trucks	2.9	0.2	1.6	0.0	0.3	5.0
Other	4.0	0.4	2.3	9.2	0.4	16.3
Subtotal, Locomotive	22.9	3.6	10.0	9.3	1.4	47.2
Fuel						48.0
Track maintenance due to locomotives						4.8
Grand Total						100.0

3.3.3 Locomotive Capital Costs

Capital costs of the individual components of a locomotive, in relation to the full selling price, were derived from prices quoted in GMD's Replacement Parts Price Book and discussions with supply industry representatives. The estimated breakdown for the base-case SD40-2 locomotive is shown in Table 3.3. The magnitudes of all of these components are sensitive to the level of plant activity and the general competitive situation within the railway industry.

TABLE 3.3

CAPITAL COST ESTIMATE FOR SD40-2 LOCOMOTIVES

	1986 $Cnd		Percent
Power plant:	$527,000		
Diesel engine		$ 408,000	24
Cooling system		68,000	4
Fuel system		51,000	3
Electrical:	$629,000		
Alternator		$ 136,000	8
PCU		51,000	3
Traction motors		340,000	20
Control		51,000	3
Other		51,000	3
Chassis:	$544,000		
Trucks		$ 221,000	13
Body and assembly		323,000	19
Total		$1,700,000	100

The diesel engine is seen to be the largest single cost component, followed by the electrical transmission package (alternator, rectifier bank and traction motors). In addition to this base purchase, there are various additional acquisition costs related to taxes, project management, quality assurance and testing.

3.3.4 Costs of Maintenance and Servicing

The costs of maintenance were derived through a three-step process. Records of maintenance activity levels were obtained from a one-year census performed by CP Rail on its SD40 locomotive fleet and a similar set of data from BC Rail. The materials costs were based on the parts prices obtained from General Motors when new replacement parts were indicated. Rebuilding costs were derived from discussions with a consultant. Labour hours were derived from BC Rail standard times for specific jobs where available and from discussion with a consultant where data did not exist. The maintenance cost analysis was disaggregated into various components as follows:

overheads
scheduled maintenance
unscheduled maintenance (on condition)
scheduled inspections/servicing
general trip servicing
failure-related costs.

In addition to the direct labour and parts costs, allowance was made for various overhead costs which can be reasonably attributed as variable with workload (as opposed to a simple allocation of fixed costs). The derivation of costs and assumptions made are discussed in Appendix A.

Locomotive Maintenance Costs versus Age. The principal source of data for maintenance activity levels was CP Rail. Data were received for all maintenance work done on CP Rail's 3,000 hp, six-axle locomotives. The data were separated of age of unit and time since overhaul in order to evaluate any sensitivity of maintenance activity to age. The age of the units varied from 20 years to less than five years as indicated below:

Date Built	Number of units
1966-70 (SD40)	65
1971-75 (SD40-2)	226
1976-80 (SD40-2)	155
1981-85 (SD40-2)	100

The average number of unscheduled shop visits and the subset of visits leading to replaced parts are illustrated for each age group in Figure 3.2. No definite correlation to age of unit exists, other than the reduced level of activity during the initial twelve months following an overhaul. This may be a consequence of the parts replacement policy for individual components. The 20-year-old units may have new major components, while more recently built units may have reconditioned components from a much older unit. Without being able to trace individual components, it is not possible to determine if there is a trend to increasing failures with age.

On the basis of the data provided, it was assumed that the history of unscheduled maintenance would repeat after each overhaul, with the activity for the first 20 percent of the cycle being one-third of the remaining 80 percent of the cycle.

It was further assumed that major rebuild (50 percent of the new locomotive price) after 20 years of service would permit an extension of the service life for a similar 20-year cycle at the same service history.

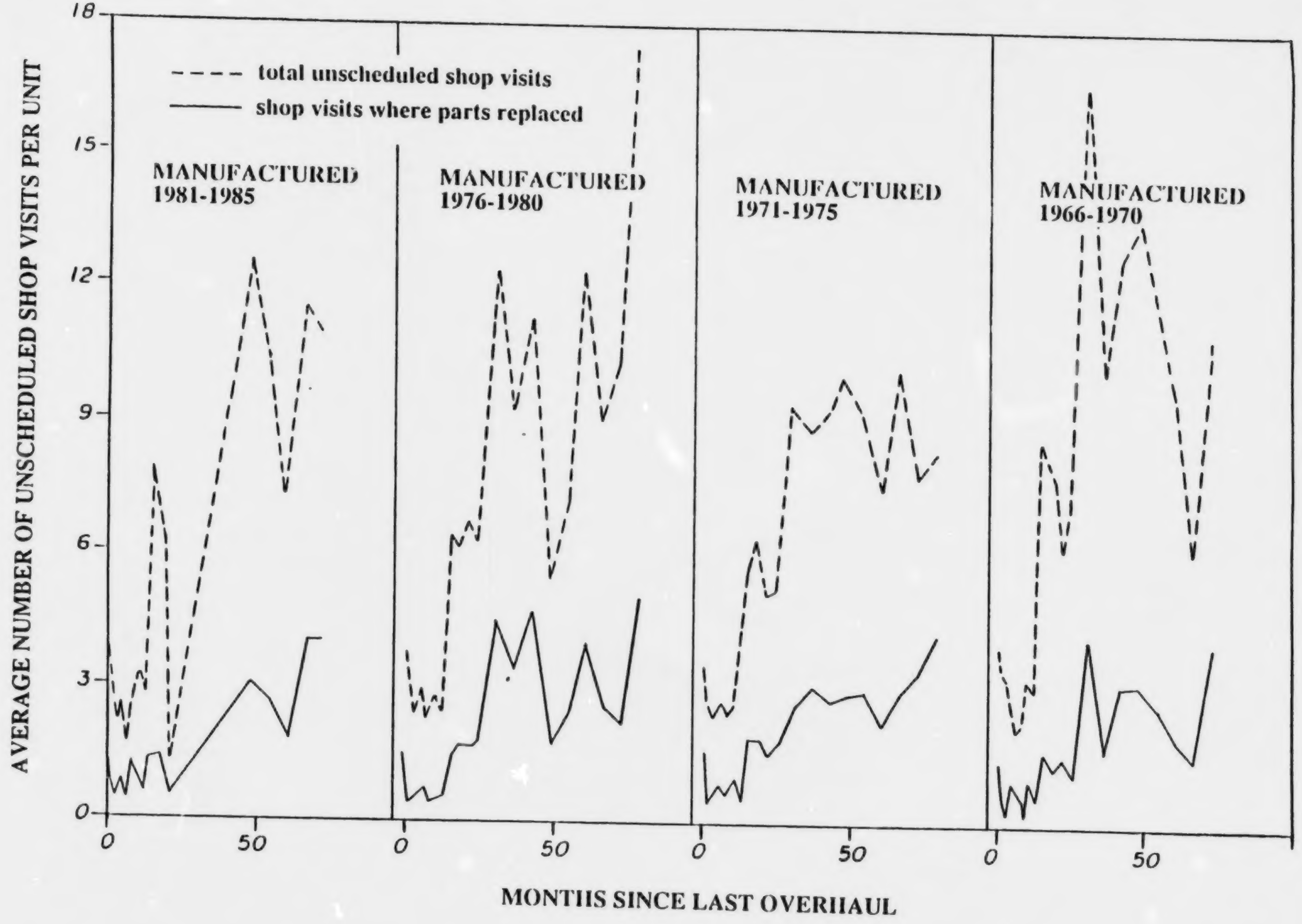


Figure 3.2 Unscheduled locomotive maintenance per unit segregated by age and time since law overhaul

3.3.5 Induced Track Maintenance

Costs of track maintenance due to locomotives were derived for the base-case locomotive as described in Appendix B. An incremental approach was taken wherein only the *incremental* costs due to the wear and tear on track attributable to the locomotive were estimated for the following categories:

rail replacement
tie replacement
switch and frog maintenance
surfacing
rail grinding.

Included in the cost estimates is allowance for the various functional overheads related to roadway maintenance. Excluded from the cost estimate is an allocation of wide-ranging corporate overheads or an allocation of total roadway-maintenance-related costs.

The costs of induced maintenance are in the range of $0.20 to $0.35 per unit-mile for a six-axle, 195-ton locomotive, depending on the track and service combination. Costs for other locomotives are scaled from this on the basis of the number of axles, weight, curving and other characteristics.

3.3.6 Fuel Costs

Fuel costs were derived from computer simulation of representative train consists over actual track profiles for CN, CP and BC Rail. These runs were used to generate "typical" duty cycles for each class of service. The duty cycle represents the distribution of time spent in each throttle notch setting. This approach permits comparison with industry-wide data on typical duty cycles. The fuel consumption in each notch setting was based on manufacturer's data for the current version of the 645 engine. Thus, the efficiency enhancements of the past decade are assumed to be incorporated into the base-case locomotive. This will produce estimates of fuel consumption lower than is the present experience but consistent with an upgrading of the engine on a rebuild.

The average hourly fuel consumption is presented in Table 3.4 for the different service duty cycles discussed in Section 2.

TABLE 3.4

PERCENT TIME ALLOCATION

Duty Cycle	Average Consumption (litres per hour)
EMD heavy	230.7
EMD medium	172.4
Bulk train service (mountain)	268.0
Express train service	251.5

Fuel costs are based on a nominal purchase price by the railway of $0.26 per litre in 1986 price levels, plus an allowance for taxes and handing. The fuel price assumption is one of the key variables in the analysis. It is also, however, subject to a high degree of uncertainty. During the last year covered by the present study, railway diesel costs fell from a reported $0.34 per litre in January to as low as $0.23 in December. While there were some increases in early 1987, prices are nowhere near 1985 levels. The analysis has been undertaken using a conservative approach to diesel fuel price increases over the next twenty years, with the real (net of inflation) price of diesel fuel returning to the 1985 high only after some 15 years, as illustrated in Figure 3.3. In adopting this conservative approach, however, it should be noted that if a motive power user expects a return of the high price increase scenario of the late 1970s and early 1980s, the relative advantage of the more fuel-efficient locomotive options will be better than shown in this report.

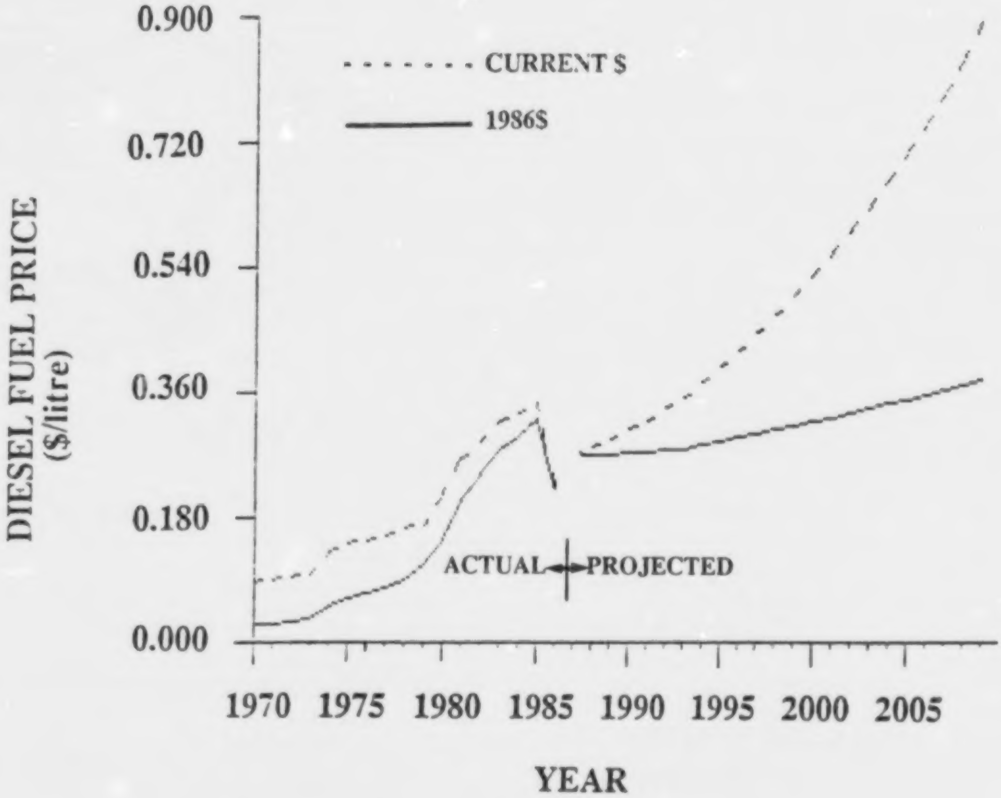


Figure 3.3 Historical and projected railway diesel fuel prices

4. VALUE OF COMPONENT IMPROVEMENTS

As shown in the previous section, fuel costs are the largest single cost factor in the full life-cycle cost (LCC) of a locomotive. Maintenance and servicing of the locomotive are second in magnitude. Those improvements which lead to reduced costs in these two areas are the focus of this section. It concerns improvements which may be considered for implementation in existing units as retrofit components. The value of some of the components considered in this section are multiplied when considered as part of the enhanced performance locomotives, as will be shown in later sections.

4.1 Diesel Engine Improvements

Projections of diesel engine efficiency improvements were made by JPL[16] in 1981. Near-term forecasts were for a 12 percent reduction in brake-specific fuel consumption (BSFC) using bottoming cycles. In the longer term, an adiabatic engine was assessed as feasible in 1995 realizing a further 11 percent efficiency gains. Little progress has been made in North America towards the implementation of either, although efficiency gains have been made.

When the JPL report was published in 1981, the conventional (no bottoming cycle) locomotive medium-speed diesel engine was considered highly efficient, with little potential for improvement. Nevertheless, small but significant decreases in full-load BSFC have occurred since then. Refinements to existing General Motors Electro-Mechanical Division (EMD) and General Electric (GE) designs, rather than any revolutionary changes, have led to these improvements.

The EMD model 645F3 engine achieved a 6.1 percent reduction in full-load BSFC between 1980 and 1983.[17] An increase in compression ratio, improved injection, modified valve timing, increased turbocharger efficiency and "fire ring" relocation contributed to the improvement. The full-load BSFC of the four-stroke GE 7FDL-16 engine has improved to a lesser degree for the same time period, as this engine was already highly fuel efficient.[18] An improved aftercooler and exhaust system, higher turbocharger efficiency, and reduced piston weight and friction were the changes applied to the GE engine.

Commonly used engines manufactured by EMD, GE and ALCO/Bombardier were originally designed when fuel consumption was less important due to the low and stable oil prices of the time. Improvements since that time have reduced fuel consumption, but it might be expected that a new design specifically intended to provide better fuel economy would offer greater potential gains than refinements to old designs. The new Caterpillar 3600 series engines were designed from the beginning for minimum fuel consumption.[19] However, in spite of this design effort, its full load BSFC does not appear to be greatly superior to the refined versions of the older engines. The full-load BSFC figures are as follows:

EMD	- 0.336
GE	- 0.328
Caterpillar	- 0.327

It can be seen that the BSFC of the Caterpillar unit differs little from the improved GE four-stroke. Thus, combustion efficiency with conventional engines appears to have peaked. Improvement trends at the level shown by EMD are, therefore, unlikely to continue through the type of refinements previously used. On the other hand, fuel consumption may improve if the efficiency can be maintained closer to the full-load figure throughout the operating range.

While BSFC is a good indication of the cycle efficiency of the power plants, improvements in fuel consumption have also been made by reducing idle speeds, eliminating parasitic losses, and implementing load-following strategies for equipment cooling requirements.

Only limited improvements can be expected from this sort of development. However, two potential areas remain -- electronic fuel injection and improved speed matching. Reductions in the order of 5 percent may be realizable with extremely small maintenance or reliability concerns. A 5 percent fuel cost saving for the base-case locomotive represents an net present value (NPV) of $167,500 per unit.

The level of R&D expenditure which such a saving would warrant is dependent on a number of considerations. Three factors are of particular importance-- the expected installed cost of the new technology, the return on investment required by the developer and the level of savings which the railway needs to retain in order to make the retrofit attractive. Table 4.1 presents the level of R&D expenditure which a 5 percent fuel savings supports under a range of these conditions. The capital purchase value is the breakeven price which would yield a hurdle level rate of return on investment of 15 percent for the railway. One might expect a much higher return on investment (ROI) to be required on operating cost savings which are dependant on fuel price forecasts and reliability of new components. The first column of Table 4.1 gives varying levels of NPV retained by the railways, while the first row shows a range of installed costs (including all overheads and profit margins). The cells of this matrix present the level of R&D expenditure which would yield a 25 percent ROI if implemented in the full fleet of 1500 mainline locomotives.

As the technology involved is principally electronics and software, one can expect relatively low installed costs but high development costs. If an installed cost of $22,000 were realizable through a $136 million development program, and if it were installed on all 1500 locomotives, the developer would realize a 25 percent ROI and the railways an NPV of $49,000,000.

Another possible improvement in fuel efficiency could be derived from changes to the thermodynamic cycle of the engine. Appendix C contains an assessment by Thermotech Engineering of the possible gains to be realized by a low heat rejection engine, and the effectiveness of a Rankine bottoming cycle. Each was assessed with a validated simulation model of Bombardier's four-stroke engine.

The steam turbine implementation of the Rankine cycle was not considered to be a viable option due to the capital and maintenance costs for the turbine and gearing or high-speed alternator. In addition, the reduction gearing would result in a significant loss in power. However conversion of some of the engine cylinders to steam was considered to offer a low cost implementation.

TABLE 4.1

FIVE PERCENT FUEL EFFICIENCY IMPROVEMENT R&D ALLOWANCE

(M$, after tax, with 25 percent ROI)

NPV of Savings (K$/locomotive, before tax)	167.50
Capital Purchase Value (K$/Locomotive, after tax)	108.80

NPV to Railway (M$)	Cost of Installed Product (K$/locomotive)					
	65	54	44	33	22	11
81.6	n/a	n/a	27	54	82	109
65.3	n/a	27	54	82	109	136
49.0	27	54	82	109	136	163
32.6	54	82	109	136	163	190

Number of Locomotives: 1,500

The steam cylinders would not present incremental maintenance costs as it would be similar to that of the diesel cylinders being displaced. Table 4.2 summarizes the power and efficiency effects of converting two of the cylinders in a four-stroke engine. Table 4.3 gives the system specifications.

TABLE 4.2

RANKINE BOTTOMING CYCLE -- FOUR-STROKE

Notch	Power Drop (percent)	Decrease in Specific Fuel Consumption (percent)
8	5	8
7	2	10
6	1	11
5	2	10
1	6	7
Idle	12	0

TABLE 4.3

SYSTEM SPECIFICATIONS, RANKINE BOTTOMING CYCLE

Intercooler:		
Steam Temperature	400°C	750°F
Steam Pressure	25 bars	375 psi
Exhaust:		
Valve Overlap	Same as existing engine	
Exhaust Temperature	593°C	1,100°F
Temperature into Turbo	333°C	630°F
Condenser:		
Temperature	120°C	248°F
Pressure	2 bars	30 psi
Heat Dissipated	1,112 kJ/s	
Approximate Size	28% of existing radiators	
Power Output	295 kW	395 hp
Steam Flow Rate	0.51 kg/s	1.12 lbs/sec
Number of Cylinders	2	
Expansion Ratio	6.1:1	

The exhaust heat exchanger should be located before the turbocharger, so that the exhaust temperature is higher, allowing a higher possible top cycle temperature, and therefore a higher cycle efficiency. In addition, for a given top temperature, the heat exchanger can be significantly smaller when the intercooler is located before the turbo.

The size of the turbo inlet would likely need to be adjusted for the reduced exhaust flow, but the size of the turbocharger itself will not need to be changed, since the same turbo is used on both 12- and 16-cylinder engines. It was not determined if the lower exhaust temperature drop entering the turbocharger would have an effect on engine performance. Turbochargers are generally not sensitive to temperature over a wide range, but it is not known if the decrease in temperature due to a bottoming cycle would be within the acceptable variation. The decreased temperatures will likely extend the life of the turbo, but this was not considered in the economic analysis.

Table 4.4 presents the range of R&D expenditures supported by the indicated savings. As opposed to the electronic systems discussed above, the steam bottoming cycle would involve a larger installed cost but a lower development cost. Four-stroke engines are in use by Canadian Railways in great numbers. The Bombardier locomotives are over ten years old. However, the relative economics apply to the purchase of new units as well as the retrofit of existing units. The table is based on a 500-unit locomotive fleet in this case. On this basis, a railway NPV of 28.2 million dollars would be realized if a unit cost of $75,000 were achieved through an R&D program of $47,000,000.

TABLE 4.4

TWO-CYLINDER BOTTOMING CYCLE
R&D ALLOWANCE
(M$, after tax, with 25 percent ROI)

NPV of Savings (K$/locomotive, before tax) 289.00
Capital Purchase Value (K$/locomotive, after tax) 187.71

NPV to Railway (M$)	Cost of Installed Product (K$/locomotive)					
	113	94	75	56	38	19
46.9	n/a	n/a	16	31	47	63
37.5	n/a	16	31	47	63	78
28.2	16	31	47	63	78	94
18.8	31	47	63	78	94	109

Number of Locomotives: 500

An active area of engine research which is directed at fuel cost reduction but not via efficiency improvements is the use of off-spec fuels. The ability of an engine to operate reliably with fuel oils which do not meet the diesel #2 specification can provide a very effective hedge against rapid fuel price escalation. During the last decade high fuel prices led to fairly rapid substitution of #6 fuel oil with lower cost coal and natural gas fuels for external combustion applications. The consequence was a surplus of #6 fuel oil which was available at 65 percent of the cost of diesel fuel. At this cost ratio, a 10 percent blend with #2 diesel realizes an effective price reduction of 3.5 percent, and a 30 percent blend achieves a saving of 10.5 percent. Both CN and CP have had many years experience with low price off-spec fuels from the Canadian tar sands. This fuel is of a low cetane (29-30 compared with 40-45 for diesel #2) but is very clean. The main concern with #6 fuel oil is that the level of impurities can vary significantly. EMD warns that engine overhaul intervals may be reduced to one-third with the use of off-spec fuels. Over half of the engine's LCC of maintenance is associated with the power assemblies (head, piston and cylinder). If these cost treble, the attractiveness of blended fuels is reduced considerably. A 20 percent blend of a 0.65 cost ratio fuel would not be attractive if it even doubled cylinder maintenance and overhaul costs. Much of the research performed to date on alternative and off-spec fuels has focussed on determining relative efficiency and overcoming combustion problems. The maintenance consequences are still uncertain. However, it is felt that the engine can be designed and, to a certain extent, modified to accommodate off-spec fuels (Table 4.5).

The Southwest Research Institute (SWRI), as a consequence of the work they have done on alternative fuels for the Association of American Railroads (AAR) and other clients, has come to the conclusion that "the diesel engine will continue to maintain its dominant position as the locomotive prime mover, irrespective of the direction taken by fuel supplies. This is because the diesel engine inherently has a strong tolerance for widely varying fuel properties, and with suitable modifications, can operate satisfactorily on a very wide range of fuels."[20]

One would expect these factors to also enhance the reliability of the engine with #2 diesel fuel. Thus, benefits would still be realized even if the fuel supply/demand balance shifted to eliminate the large discount for #6 fuel. The value of these engine improvements would be the same as those represented in Table 4.1 above for a 5 percent efficiency improvement if they permitted a 14 percent blend without increasing engine maintenance costs. The minimum level of benefits are those associated with reduced maintenance costs and continued use of #2 diesel fuel. Table 4.6 presents the relative benefits of reducing the unscheduled maintenance of the locomotive by two-thirds. Overhauls would be at the same five-year interval but would have a higher parts cost. Column 3 of the table allows about a 30 percent premium on existing power assemblies.

4.2 Traction Motor Improvements

The series-wound dc motor was the choice for early railway traction applications because its natural characteristics were well suited to the requirements. Its torque-speed characteristics provide good natural load-sharing characteristics when connected in either parallel or series. Early diesel-electric locomotives used a dc generator. In order to reduce current requirements, the traction motors were connected with pairs in series when in the low-speed/high-torque region. In order to reduce the voltage requirements at high speeds, the motors were connected in parallel and field weakening bypass resistors were cut in. This method of operation realized a low equipment cost but presented operating performance problems. The series connection of motors produced a poor slip-recovery characteristic and field weakening led to communication problems and flashovers of the dc motors.

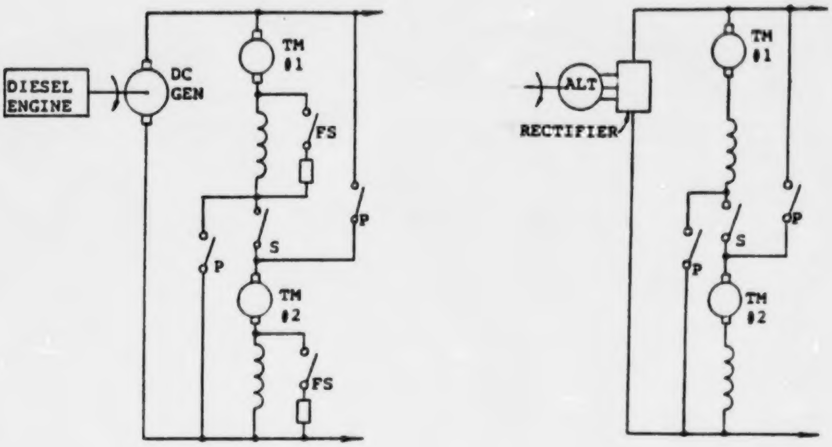
Evolutionary improvements in the performance of the electric transmission of North American locomotives have also been achieved. Hapeman et al.[21] have documented the development to date. This evolution in development is summarized in Figures 4.1(a) through 4.1(d). The availability of high power semiconductors led to the replacement of the dc generator with an ac synchronous generator-rectifier set (Figure 4.1(b)) It also allowed high voltages to be applied in place of field weakening at high speeds. The next step was to overcome the low-speed slip control problems of the series-parallel motor connection by moving the series-parallel transition from the motors to the alternator windings (Figure 4.1(c)). The final step made on some locomotives was to remove transition completely (Figure 4.1(d)).

TABLE 4.5

HEAVY PETROLEUM FUELS: PROBLEMS AND REMEDIES

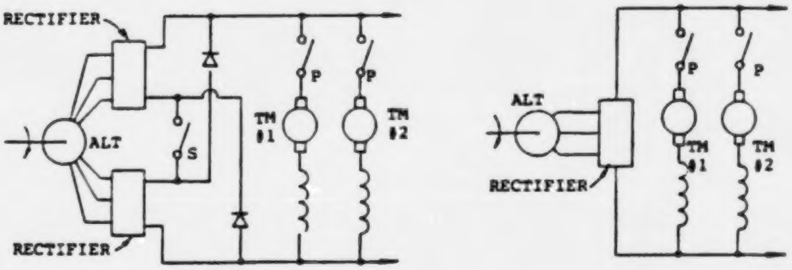
Fuel Property	Engine Problems	Remedies
High Vanadium content	Piston corrosion at surface temperatures above 1100-1200°F	Cool piston by oil or water
High viscosity	Fuel pumping difficult	Heat fuel to reduce viscosity
	Injector nozzle deposits	Cool nozzle, reduce SAC volume
	Piston deposits	More complete combustion by reducing engine speeds, warming surfaces, using higher loads. Prevent fuel impingement on surfaces.
High carbon residue	Deposits in two-stroke ports.	More complete combustion. Cool port surfaces.
	Poppet valve sticking	
	Poppet valve seat damage	Use special lubricants
	Ring fouling	More complete combustion. Cool ring belt. Use special lubricants.
	Wear	More complete combustion. Use abrasive-resistant materials.
High ash content	Wear	Use abrasive-resistant materials.
High sulphur content	Corrosion	Maintain surfaces above dew-point temperature. Use corrosion-resistant materials. Use high-alkalinity lubricants. Prevent blow-by access to crankcase.
Contaminants	Wear	Clean fuel by filtering/ centrifuging
Low cetane number	Knock, incomplete combustion.	Increase engine load. Reduce engine speed. Reduce ignition delay by pilot injection, or by equivalent method.

Source: C.D. Wood, "Alternative Diesel Fuel Research," *Proceedings, Railroad Energy Technology: The Alternatives*, Association of American Railroads, 1983.


DIESEL ENGINE
DC GEN
TM #1
FS
P
S
P
TM #2
FS
ALT
RECTIFIER
TM #1
P
P
S
TM #2

(a) DC generator plus motor transition (motor field shunting)

(b) Alternator rectifier plus motor transition (no motor field shunting)


RECTIFIER
ALT
S
P
P
TM #1
TM #2
RECTIFIER
ALT
RECTIFIER
P
P
TM #1
TM #2

(c) Alternator rectifier transition, motors in parallel

(d) No transition motors in parallel

Source: M.J. Hapeman et al., "Diesel Electric Locomotive Propulsion Systems, A Look Into the Future," IEEE, CH2167-5/85/0000-0108, 1985

Figures 4.1 Evolution of motor control

TABLE 4.6

REDUCE UNSCHEDULED CYLINDER MAINTENANCE BY TWO-THIRDS
R&D ALLOWANCE
(M$, after tax, with 25 percent ROI)

NPV of Savings (K$/locomotive, before tax)	94.67
Capital Purchase Value (K$/locomotive, after tax)	61.49

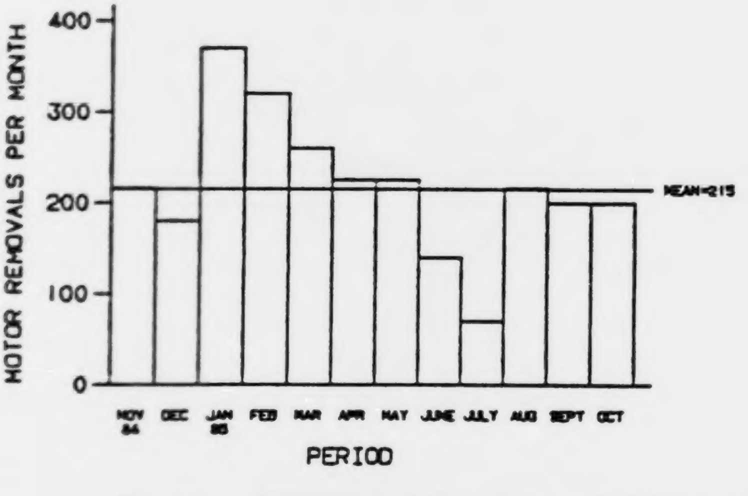
NPV to Railway (M$)	Cost of Installed Product (K$/locomotive)					
	37	31	25	18	12	6
46.1	n/a	n/a	15	31	46	61
36.9	n/a	15	31	46	61	77
27.7	15	31	46	61	77	92
18.4	31	46	61	77	92	108

Number of locomotives: 1,500

The base-case locomotive (SD40-2) is represented by Figure 4.1(b). In comparison to later-generation locomotives, one can expect higher maintenance and reliability costs associated with transition control and poorer adhesion performance at low speeds. Current generation locomotives also have an upgraded traction motor. Improved electrical insulation materials, additional field windings and better heat transfer design have led to higher continuous current ratings and thus higher available torque. An important parameter in the design of locomotive transmission systems is the utilization of available friction at the interface of wheel and rail. This adhesion performance of a locomotive is considered to be the ratio of tractive effort to gross weight at which a train can be dispatched with a reasonable level of confidence that it will climb the steepest grade on the route. As such, it contains subjective opinion and can be expected to vary for different railways and for different lengths of grade. CN Rail dispatchs SD40 locomotives at 19 percent adhesion and increases this to 22 percent if the unit is equipped with their add-on "Positrack" system of wheel slip control. CP, on the other hand, dispatches its SD40 at 18 percent. Their limitation is imposed on the basis of traction motor failure experience as well as grade-climbing ability. As noted in Section 2, their gradient conditions are much more severe than those of CN. Maximum tractive efforts are required for continuous intervals of up to 60 minutes in the principal bulk haul region. In CN's case, these times are almost an order or magnitude smaller.

Traction motor maintenance has been found to account for 36 percent of a locomotive's life-cycle maintenance cost. Both CP and CN have been concerned with the high failure rates of dc traction motors. Traction motor failures are seasonal, with snow ingestion considered a major cause of winter failures. The experience of CP Rail has been documented by Cavanaugh et.al.[22] Seasonality trends, as illustrated in Figure 4.2, indicate that there is about a 14 percent increase in failures over the winter months. The system average mean time between failure is 31 months and the average cost of repair is $5,700. However, the full cost of maintenance for mainline 3000 hp locomotives in our sample is higher. The $5,700 figure considers direct costs only and does not include the small percentage of new replacement motors or the field labour in changing out the motor. Also, the subset of high-horsepower mainline locomotives included in our data analysis experienced close to a 60 percent failure rate or a mean time between failures (MTBF) of 20 months.

The value of deriving a better snow resistant performance is assessed in Table 4.7. The table is based on only the value of implementing snow protection measures, since the problems of winter failures has been investigated at CN with good reported success. After a number of years of investigation they implemented design changes to the body, and both the intake and discharge sides of the vent system to combat snow related failures.[23] Their GP-9s are being remanufactured with these changes and all new locomotives are wide-bodied with the "Draper Taper" modification to improve rear visibility. The benefits to the railway would be an NPV per locomotive of $25,890, minus the incremental cost of the retrofit or price of new locomotive.



Source: Cavanaugh, S.E., et al., "Freight Locomotive AC Traction Technology in North America," *Third International Heavy Haul Railroad Conference*, October 1986.

Figure 4.2 Traction motor removals (CP Rail road locomotives, 11/84-10/85)

TABLE 4.7

PROTECTION OF DC TRACTION MOTOR

NPV of Savings (K$/locomotive, before tax)	39.86
Capital Purchase Value (K$/locomotive, after tax)	25.89

Research in dc traction motors includes investigations of a slotless armature winding by British Rail and U.S. Navy funded research into long life, high current density brush/commutator materials at the University of Virginia. The slotless armature achieves much lower commutation voltages by placing the armature windings on the surface of the iron core rather than wedging them into slots. However, as discussed in Appendix D, the likelihood of achieving an effective slotless dc traction motor are very low. Also, in the Canadian railway application, commutation problems are only a minor part of traction motor maintenance expense.

Similarly, the development of a long life brush does not alleviate the major failure problems of the dc traction motor, only its scheduled maintenance costs. On the other hand, the realization of a higher current density could permit a reduced commutator length and thereby an increase in the active armature length for the existing dc traction motor volume. The University of Virginia research is in the early stages of materials investigation to find a suitable commutator material for a wire filament brush already tested.[24]

AC Traction Motors. The combined benefits of these developments and more can be achieved with ac traction motors. Appendix D presents a detailed comparison of the inherent performance characteristics of ac and dc traction motors. AC traction has been employed extensively in Europe, where service applications, which are largely power (rather than tractive effort) based, realize the full benefits of the motor's inherent advantages. While the dc traction motor is limited in rotational speed by the surface speed of its commutator, the ac traction motor can be operated at 50 percent higher speeds. Thus the power-to-weight ratio of the motor is much better. A lightweight traction motor realizes reduced fuel and track maintenance costs in high-speed passenger service.

This benefit is mitigated in high-tractive-effort applications where the motor must be designed for high torque rather than high power. The torque-to-weight ratio of an ac traction motor is only slightly higher than the dc traction motor. Yet, there are still benefits to be derived. An ac traction motor with only a small weight saving in a torque-constrained application, will provide a much wider constant power speed range (50 to 60 percent). Thus, where the dc traction motor requires different gearing to run in European freight and passenger applications, the ac traction motor allows the same locomotive to be used in both services. It is this universality which provides much of the economic justification for widespread use of ac traction in Germany, where a 20 percent reduction in locomotive units is expected with ac traction.[25]

France has also found ac traction to be economical because of its universality.[26] In their case, the ac synchronous motor has been chosen.

The economics of ac traction are different in Canada. The spread in service applications is much smaller. Passenger locomotives are not operated by the major railways and the existing railway-imposed tractive effort requirements of bulk freight operations and top speeds of express locomotives can be met with dc traction motors in most areas. Later sections of this report highlight the importance of a wide service application in relation to much higher tractive effort performance. This section assesses the value of ac traction as a retrofit to the base-case locomotive at its performance level.

The principal benefits of ac traction in existing Canadian railway practice would be derived from its much reduced maintenance costs. Without a commutator and risers, which are a maintenance factor and also susceptible to moisture-related faults, the ac traction motor realizes savings in both scheduled and unscheduled maintenance. However the elimination of the commutator cannot be fully utilized by increasing the active torque-generating length of the ac traction motor, since it requires almost equivalent length for the end turns of the stator windings.

The squirrel cage rotor costs significantly less than the dc motor armature and, being uninsulated, is not susceptible to moisture grounding faults and deterioration of insulation from overheating. The stator holds the insulated current-carrying conductors at line voltage, with two advantages. It has a larger circumference than the armature and can therefore provide more cross-sectional area per current-carrying conductor. The stator provides a larger surface area for heat transfer and is more readily accessible for liquid cooling if desired.

Reliability. While the cost of motor maintenance is reduced significantly with ac traction, the addition of many new power electronic and control electronic components leads to reliability concerns.

The railways using ac traction have been monitoring the reliability of these units. Since they are all in the prototype stage, the number of early failures was high in each case but has steadily improved with relatively simple design modifications.

Braun[27] has reported MTBF of 540 days for power semiconductors and 1080 days for control electronics. Early problems with commutating capacitors and torque pulsations in the traction motors have been mitigated. In 1982, the MTBF for the complete locomotive was 270 days. These are shunting locomotives which see a utilization rate about the same as our base-case locomotives (50 percent) but the milage is much lower.

The Norwegian State Railways (NSB)[28] have realized a twofold improvement in reliability of their six electric EL17 locomotives, from 15,000 km between defects in 1983.

The German Federal Railways (DB) undertook extensive acceptance testing of their five model E 120, three-phase electric locomotives.[29,30,31] The units were tested for extended periods in a cold chamber at -30°C and in a baking room at 64°C. The prototype units underwent four redesign stages -- power hardware and software, and auxiliary hardware and software. Reliability has improved significantly with each retrofit. Prior to the last retrofit, the maintenance experience was 3.5 man-hours/1000 km (MH/Mm) of which 1.1 was scheduled). There was an average of 2.3 failures per month per locomotive in the complete electric package. With the retrofit, maintenance decreased to 2.1 MH/Mm. The long-term forecast for production units is 0.55. These figures are for the complete locomotive. Forty percent of initial failures were of the mechanical or conventional electrical equipment. The projected maintenance cost for the new electrical components is 0.43 man-hours/1000 km. In comparison to the estimates for the base-case locomotive, this would be a 240 percent increase in electrical control equipment maintenance.

The maintenance savings estimated for ac traction are as follows:

	Motor	PCU & Control
scheduled maintenance	- 90	+ 240
overhauls (five-year frequency)	- 50	0
inspections	- 90	+ 240
unscheduled maintenance	- 62	+ 240
road failures	- 62	+ 240

The findings of a survey of failure causes undertaken in the summer months was provided to the study team by CP Rail. The seasonally adjusted rates and the underlying assumptions for the reduction in motor failures and unscheduled maintenance are shown in Table 4.8. The failure comparison is based on ac traction motors eliminating all snow- and commutator-related failures, half of the overheating and grounded armature-winding failures, and all field pole ground failures. These estimates may be somewhat conservative as the average cost of an overhaul was applied to the reduced number of failures. In fact, those categories of failure that are not affected (pinion and external leads) are relatively small in comparison to those which are affected (grounded coils and commutator problems). The loss in reliability of the more complicated electronic power system may also be offset by the improvement realized by fewer electro-mechanical relays and contacts used in the base-case SD40 through ongoing reductions in the number of semiconductors (SCRs) required in the inverters (the CP prototype has two SCRs per schematic device, while the E 120 has four).

The life-cycle values of operating cost savings derived for these assumptions are given in Table 4.9.

The capital purchase value is about one half of the budgetary price estimate received for a pulse width modulated (PWM) voltage source converter in this application. An expected 20 percent cost savings in the motor and alternator is not enough to make up the difference. However, on the basis of cost details provided by French,[32] for a PWM inverter, about 40 percent of the cost is overhead contributions. These can be expected to be reduced with volume. Also, as will be discussed later, these benefits are multiplied if ac technology is more effective in enhancing locomotive performance and versatility.

TABLE 4.8

SEASONALLY ADJUSTED MOTOR FAILURE RATES

Failure Mode	DC	Three-phase Failure Multiple	Three-Phase
Snow	0.14	0.0	0.00
Pinion	0.07	1.0	0.07
Grounded Armature	0.12	0.5	0.06
Commutator	0.07	0.0	0.00
Over-heated Armature	0.09	0.5	0.045
String Bands	0.05	0.0	0.00
External Leads	0.06	1.0	0.06
Grounded Field Coils	0.05	0.0	0.00
Total (known failures)	0.67		0.235
Overhaul	0.08	0.50	0.04
Other	0.25	?	?
Three-phase/DC Failure Ratio			0.35

Source: CP Rail

TABLE 4.9

AC TRACTION MOTOR AT SD40 PERFORMANCE

NPV of Savings (K$/locomotive)	326.50
Capital Purchase Value (K$/locomotive, after tax)	212.07

The economic evaluation of the competitive ac traction motor alternatives is beyond the scope of this study. The major factor in such an assessment is the price of the hardware involved, and this is driven as much by marketing factors as it is by technological differences. The ac drives presently in use include both current-source and voltage-source inverter-supplied induction motors and line-commutated synchronous motors.

The synchronous motor realizes cost advantages in the converter design because its separately excited field can be used to commutate the SCRs, whereas the inverters of ac induction motors require expensive forced commutation circuitry. While the use of gate-turn-off thyristors (GTOs) would eliminate the need for this extra circuitry with induction motors, their development has not yet reached the capacity/price ratio necessary to compete with SCRs in high-current, high-power inverter applications. Mitigating the synchronous motor's advantage in the inverter cost is the added first cost and maintenance cost of the motor.

These costs are associated with the separately excited field winding on the rotor -- the same characteristic which lowers the inverter cost. The brush gear and slip rings which feed the dc supply to the field winding are not as significant a cost item or maintenance concern as the dc traction motors commutator. The main concern in the Canadian application is that the rotor requires an insulated winding and is thus subject to grounding and snow-related failures, as is the dc motor's armature. These concerns would be removed if a permanent magnet field were used with the synchronous motor. Such motors are common at low power levels but field strength limitations of permanent magnets and the benefits of a controllable field have prevented their application to high-power motors. As discussed in Appendix D, the development of high magnetic field density permanent magnets may permit a permanent magnet ac synchronous traction motor.

Truck Improvements. Benefits of improved truck design are largely associated with the adhesion performance which it permits. This aspect is discussed in later sections. At this point, the benefits of truck improvements to the base-case locomotive will be assessed. In this respect, curving performance is the principal attribute affecting operating costs. Track maintenance, frictional dissipation of energy and wheel wear need to be considered. Rigid-frame two- and three-axle trucks are used in Canadian railway operations. Interconnected rigid-frame trucks are used extensively by the Swiss Federal Railways[33] on high-curvature routes. Prototype forced steering trucks, which have been given flange-free curving up to 5° curvature (350 m), are being evaluated by British Rail.[34]

The benefits of improved curving performance would be highest in the mountain regions where over 50 percent of the track is curved. In this region the three-axle truck is the standard for all three railways. The relative curving performance of a number of trucks was investigated with a computer simulation model developed by C. Fortin and R. Anderson. The model and simulation results are presented in Appendices E and F.

The corresponding peak and average flange forces for a number of trucks is compared in Table 4.10.

TABLE 4.10

PREDICTED FLANGE FORCE COMPARISON

		Six-axle Locomotives		Four-axle Locomotives	
Degree of Curve	Stiff Centre Axles	Centre Axle with 24 mm Lateral Freedom	Inter-connected Trucks	Standard Trucks	Inter-con. Trucks
4° lead axle flange force kN)	57	44	25	36	18
avg. HR flange force (kN)	16	22	8	13	5
lead axle angle of attack (°)	-.51	-.57	-.42	-.44	-.31
8° lead axle flange force (kN)	64	38	18	35	14
avg. HR flange force (kN)	17	22	7	16	4
lead axle angle of attack (°)	-.90	-.84	-.73	-.81	-.64

While the rail wear benefits are derived from every curve on the route, fuel consumption benefits are derived only from those curves where tractive effort is required. On the basis of 0.8 lb/ton per degree of curvature, fuel consumption due to curve negotiation has been estimated to be 150 litres for the six-axle base-case locomotive. With gauge-face lubrication, curve resistance is reduced by 50 percent. On the basis of lubrication on curves over 2 degrees, the consumption is reduced to 77 litres. The frictional forces involved in curve negotiation are the source of each of the cost factors of interest. The NPV of reducing these forces by 40 percent and 90 percent are shown in Table 4.11.

The R&D value associated with the 40 percent reduction is shown in Table 4.12. The number of locomotives which the retrofit would be applicable to are an estimated 400 in dedicated mountain region service. Column #3 of the table represents a 10 percent increase in the price of the base-case truck. All numbers in the table are doubled for the case of an 80 percent reduction. An allowance for incremental truck maintenance has not been included in these values. If truck maintenance increased by 20 percent, the values in the table would be reduced by 8 percent. The contribution of improved curving performance to adhesion performance has not been accounted for but is discussed in a later section.

To this point the benefits of component developments directed at lowering specific cost elements of the base case-locomotive have been considered. The most significant savings in locomotive LCC can be derived from enhancing performance such that a complete unit may be displaced. It is this level of performance which is the prime consideration in a purchase-versus-rebuild decision and for new locomotives. The following section evaluates the value of performance enhancement in terms of tractive effort and power for the Canadian railway situation.

TABLE 4.11

COST SAVINGS FROM TRUCK CURVING IMPROVEMENTS (NPV K$)

	Reduction in Angle of Attack	
Cost Component	40 percent	80 percent
Track Maintenance	43.5	87.0
Fuel Consumption	7.8	15.6
Wheel Maintenance	18.6	37.2
Availability Effect	16.5	33.0
Total	86.4	172.8

TABLE 4.12

TRUCK CURVING ANGLE OF ATTACK REDUCED 40 PERCENT R&D ALLOWANCE (M$, after tax, with 25 percent ROI)

NPV of Savings (K$/Locomotive, before tax)	86.40
Capital Purchase Value (K$/Locomotive, after tax)	56.12

	Cost of Installed Product (K$/locomotive)					
NPV to Railway (M$)	45	28	22	17	11	6
11.2	n/a	n/a	4	7	11	15
9.0	n/a	4	7	11	15	19
4.5	n/a	11	15	19	22	26
2.2	4	15	19	22	26	30

Number of Locomotives: 400

5. VALUE OF LOCOMOTIVE PERFORMANCE ENHANCEMENT

5.1 Physical Performance Characteristics

Locomotives are allocated to linehaul service on the basis of one of two criteria -- either sufficient power and speed to achieve the desired run times, or sufficient tractive effort to climb the steepest grade on the route. Thus, tractive power and tractive effort are the two most important characteristics of a locomotive. These characteristics are illustrated in Figure 5.1.

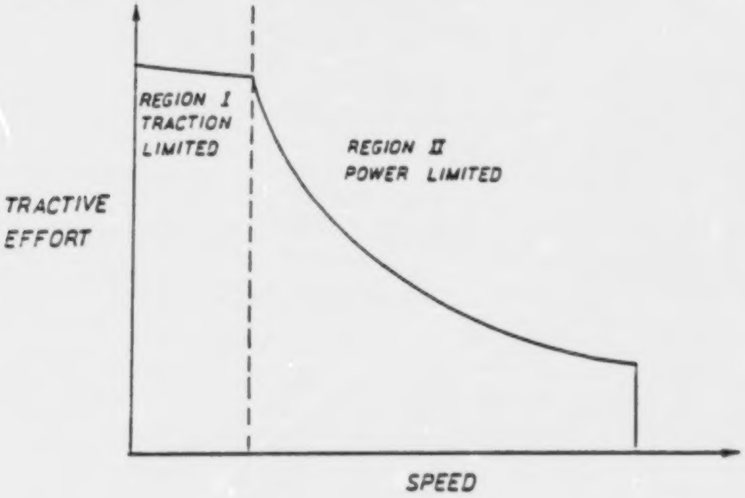


Figure 5.1 Locomotive tractive characteristics

Region II is the power-limited region and represents the primary operating region for the locomotive. The available tractive effort in this region is limited to:

$$TE(v) = k\, g\, e(v)\, \frac{(P_{max} - P_{aux})}{V} \qquad (5.1)$$

where: TE = tractive effort as a function of speed

P_{max} = maximum engine power

P_{aux} = engine power required for auxiliaries

$e(v)$ = efficiency of power transmission from engine to the wheel/rail interface as a function of speed

g = gear ratio

k = units conversion factor.

From equation 5.1, it is obvious that the tractive effort in this operating zone is limited by the maximum engine power, the auxiliaries requirement and the efficiency of transmitting the traction power from engine to wheel. For the diesel-electric locomotive, engine power is limited by the weight restrictions of the track. In this regard, Canadian railways impose a limit of 65,000 lb/axle. Many U.S. railways permit 70,000 lb/axle for locomotives while maintaining the 65,000 lb/ axle limit for rolling stock. Auxiliary requirements are dominated by the cooling requirements of the engine and traction motors. Much of the improvement in tractive power and fuel efficiency of the current-generation of locomotives has come from a reduction in these auxiliary loads by implementing a load-following operating strategy. The efficiency of transmission is tied to the technology employed. The dc series-wound traction motor is used in past and present generations of North American diesel-electric locomotives.

It is the transmission technology that limits the width of Region II. The role of power transmission in a railway application with an on-board power source is demanding. Adhesion limitations require a distribution of prime mover torque to several axles, while the economies of scale for diesel engines necessitates a single prime mover. In matching the performance characteristics of engine and transmission, compromises must be made. With fuel costs and engine maintenance very high considerations in the system design, the function of the transmission is to allow the engine to operate at its most efficient speed for a given power demand. In North America, this requirement for a continuously variable transmission is met with an ac generator-rectifier-dc motor combination. The ideal transmission would extend the boundaries of both axes of Figure 5.1. The effectiveness of the transmission may be measured by the extent to which it limits the boundaries of both the tractive effort and the speed axes.

The tractive effort for a self-contained locomotive is also limited by the adhesion available between wheel and rail. Adhesion permitting, the power output of the engine is limited at low speeds by the torque conversion limitations of the transmission. For an electric transmission, this is governed by the current limit of either the generator or the traction motors. At high speeds, voltage limits, rotational speed limits, and field weakening limits are the constraints.

Within limits, the gearing ratio of equation 5.1 provides a scaling factor for the two axes of Figure 5.1, and thus some flexibility in the location of the two boundaries of the constant power region. Higher tractive effort may be achieved at the expense of maximum speed. Figures 5.2 (a) and (b) present the tractive effort/speed curves for the base-case SD40-2 and current-generation SD60 locomotives.

5.2 Tractive Effort Value Estimates

For bulk operations, the limitation to the reduction of locomotive units is primarily tractive effort. Bulk operations often involve annual quotas where the direct time value of individual shipments is not an important factor. Trains are dispatched with the minimum power and tractive effort necessary to make the ruling grade. An increase in the tractive effort capability of a locomotive can be translated into reduced units. The lower-powered train reduces locomotive unit-kilometers and, in operating at a lower power-to-weight ratio, also realizes fuel savings. The penalty associated with the lower power-to-weight ratio is an increase in transit time. While the direct impact on commodity delivery is not important in this application, there are several other impacts. The investment in train sets and cars must be increased to accommodate a higher cycle time. Also, the interference effects of reduced train speeds on other trains and on line capacity are important considerations.

Figure 5.3 presents direct operating cost savings from increased tractive effort performance for a representative bulk unit train service in Canadian mountain conditions. The savings are in relation to the operations performance of base-case SD40 locomotives. The alternatives are based on presently available engine powers (2,865 kW) and assume the same unit cost performance as the SD40. However, all but the first involve enhanced tractive effort performance. The net direct operations cost savings of higher tractive effort performance are, at this point, based on unchanged maintenance costs per unit-km.

The first case represents the replacement of the base case SD40s with current-generation 3,800 tractive hp locomotives dispatched at the same power-to-weight ratio as the base case. The required dispatch-adhesion performance is 23 percent. The savings result from the efficiency, power and tractive effort enhancements of current-generation locomotives. Table 5.1 presents the savings for each of the four cases in terms of break-even purchase price for a locomotive of the corresponding performance. Two values are given. The first represents the indifference price in a buy new comparison with the base case locomotive priced at $1.7 million. It is obvious that the current-generation locomotive will achieve significant economies over the base-case SD40, if it exhibits the same maintenance and reliability performance as its predecessor. The second value represents the indifference price in a rebuild comparison where an $800,000 rebuild is assumed to achieve the fuel economies of the improved 645 engine and the same 20-year maintenance cost experienced in past 20 years. In addition to the cost of the rebuild, there is an estimated opportunity cost of $200,000 associated with the salvage value of the locomotive components if it is not rebuilt. In this situation, the indifference price is below the equivalent $/hp ratio of the SD40 (i.e. 3,800 * ($1.7M/3,000) = $2.15 M).

Taking, for example, the 0.252 kN/kW traction performance ratio, this optimistic determination of savings means that a 2,865 kW locomotive which provides 745 kN of tractive effort is economical at up to an incremental life-cycle cost per locomotive of $6.3 million. Thus, there is a significant margin for incremental operating costs and front-end development costs.

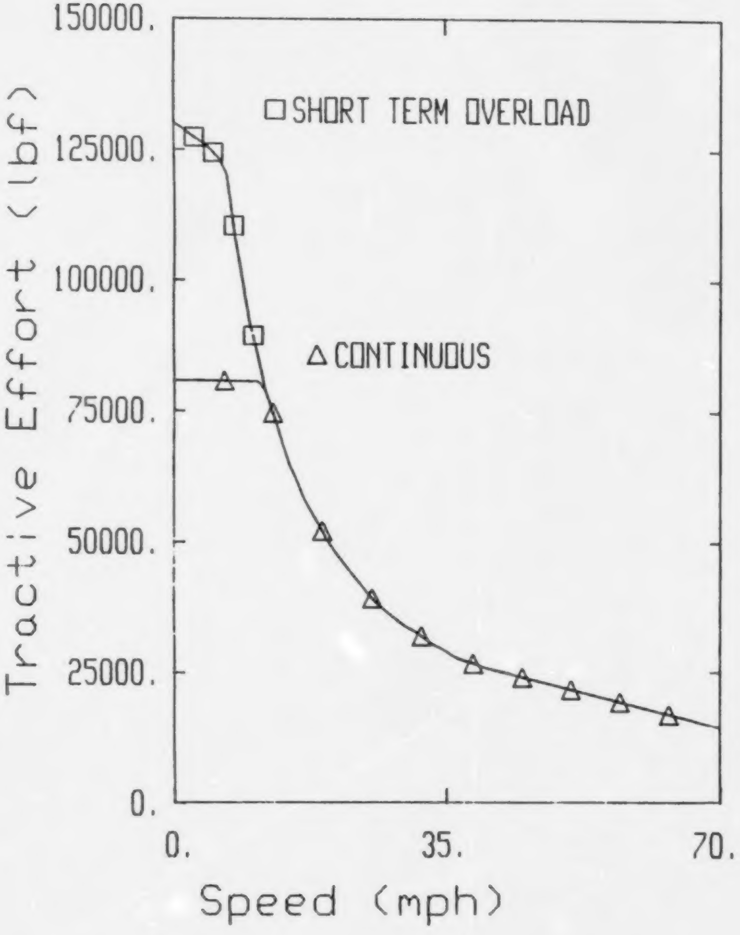


Figure 5.2 (a) SD40-2 tractive effort

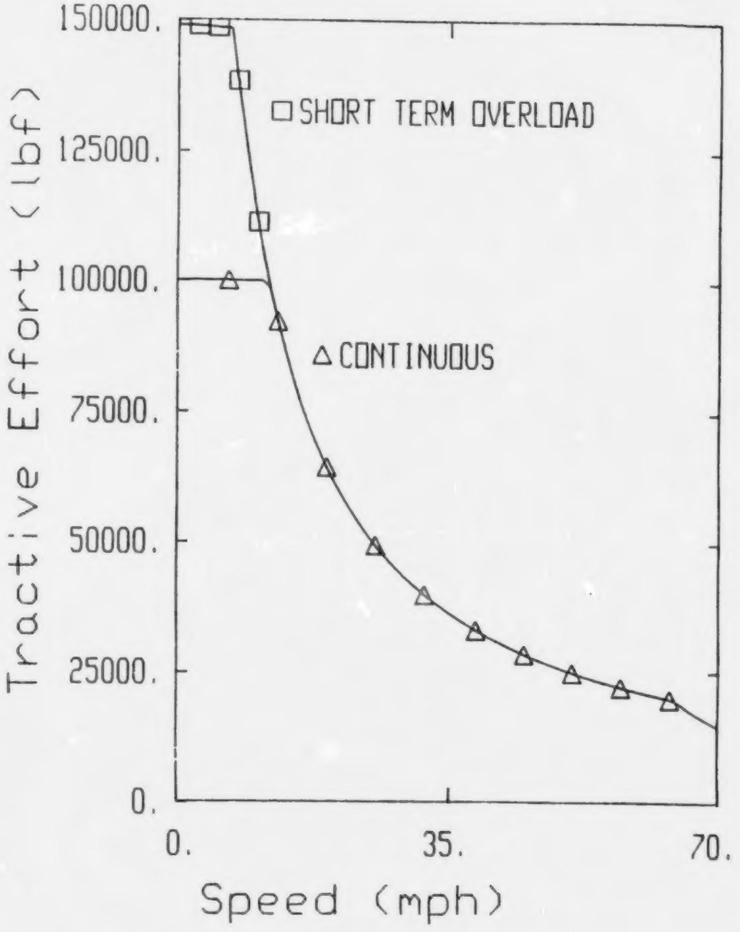


Figure 5.2 (b) SD60 tractive effort

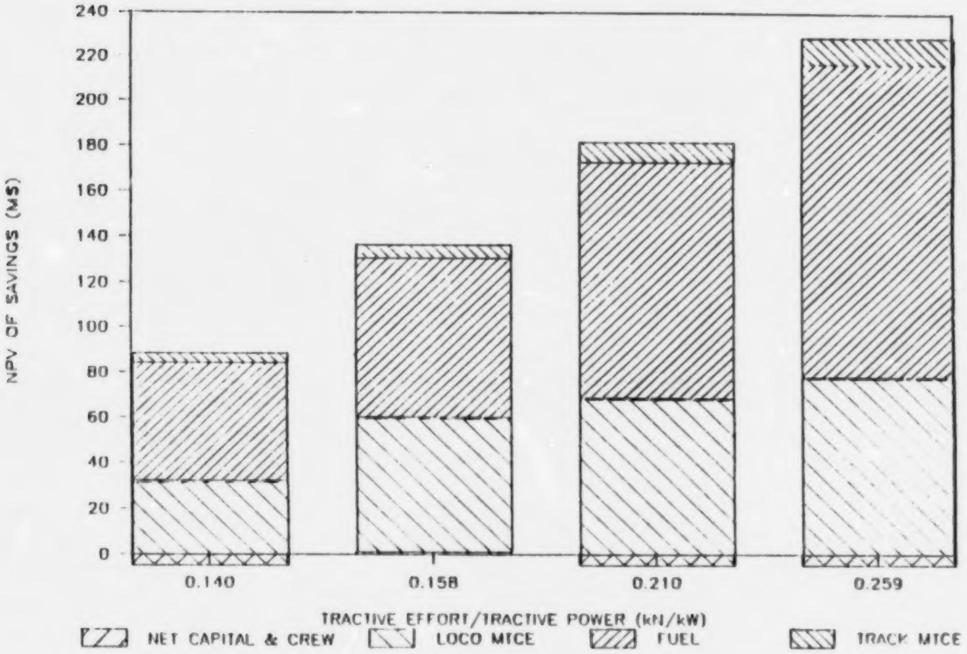


Figure 5.3 NPV of operations savings for hypothetical bulk commodity operation (based on unchanged unit costs per locomotive)

TABLE 5.1

VALUE OF ENHANCED TRACTIVE EFFORT FOR A 2,865 kW LOCOMOTIVE
(M$, after tax)

Tractive Effort/Power Ratio (kN/kW)	0.140	0.158	0.210	0.252
Buy New Comparison	2.912	3.811	5.205	6.28
Rebuilt Comparison	1.988	2.766	3.894	4.71

5.2.1 Other Cost Factors

Negative unit cost and indirect cost impacts are not as readily determined. The economic impacts on infrastructure capacity are very site-specific and will change with traffic growth over the life of the locomotive. Two factors are important in this regard -- the incremental interference with other trains, and the time it takes to climb the steepest grade on the line. The impact on operations for CP Rail's mountain line was analyzed from these viewpoints using a line capacity computer model.[35] This line was chosen because of its severe gradient conditions, its high traffic density, and because it has been the subject of other motive power evaluations in the past.

The influence of a reduced power-to-weight dispatch ratio for coal unit trains was evaluated. It was found that for existing traffic levels, the impact on other traffic can be minimized. This is a consequence of ongoing infrastructure investments made by CP to increase capacity, largely by double-tracking critical gradient segments. With traffic growth, the lower power-to-weight operation would run into capacity problems first. Figure 5.4 illustrates this impact in terms of available growth capacity. Growth capacity drops significantly with higher traction ratios.

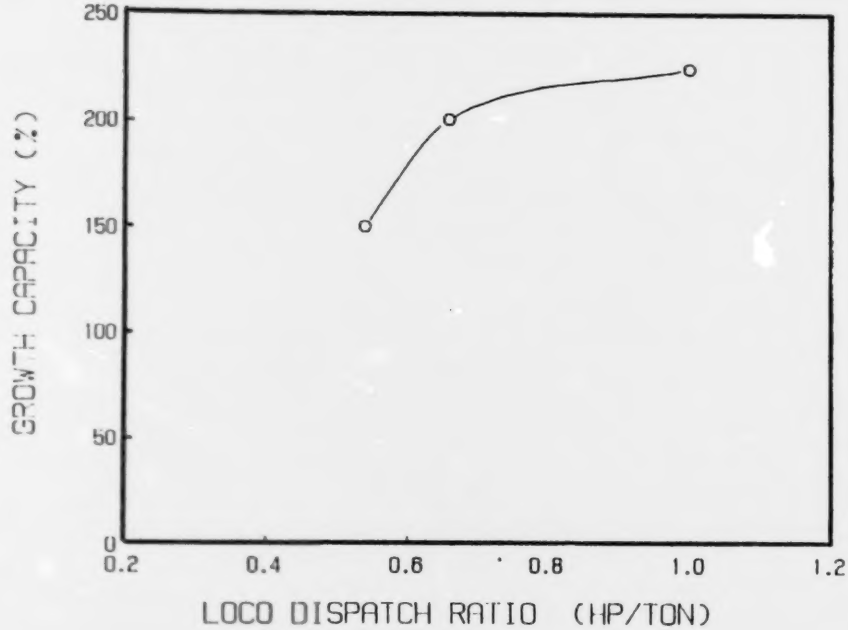


Figure 5.4 Infrastructure capacity impacts

The time to climb the steepest grade on the route increases rapidly at dispatch ratios above 0.18 kN/kW, as illustrated in Figure 5.5. Here again, the investment in double track on this and other steep grades alleviates some of the past concern for this factor. Figures 5.6(a), (b) and (c) compare the transit time performance of each of the four trains on the steepest grade on the line. The situation represents peak traffic conditions and a six-hour maintenance window. Under these conditions the priority of some trains had to be adjusted so that the 0.265 kW/kW case would be feasible. Its negative performance impacts are noticeable in comparison to the other two cases. Given the apparent knee in the capacity influences and the diminishing operating cost reductions between 0.66 and 0.54 hp/ton dispatch ratios, the 0.54 hp/ton case (i.e. the 0.264 kN/kW performance ratio locomotive) was dropped from further consideration.

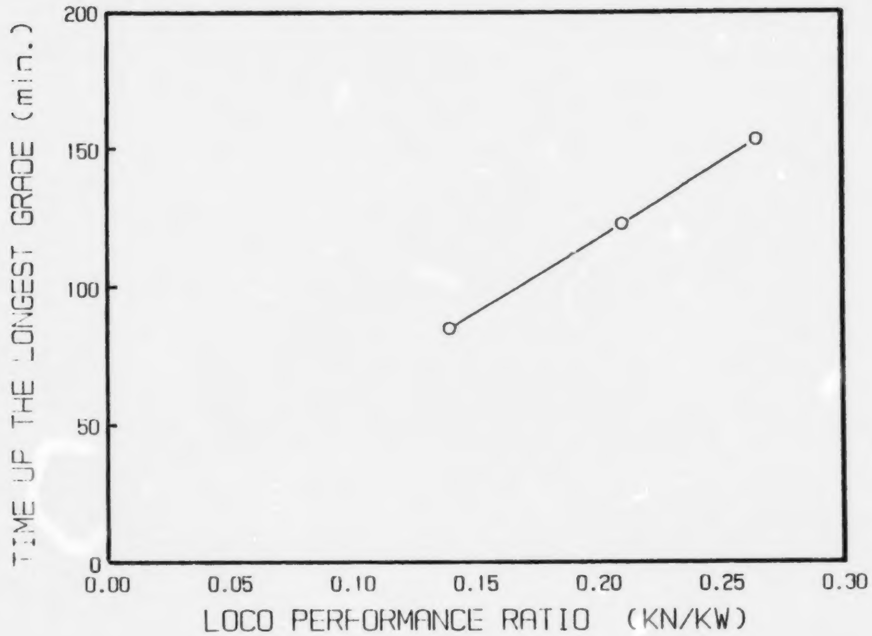


Figure 5.5 Transit time impacts

The benefits of enhanced tractive effort performance have been demonstrated. Section 6 looks at some of the hurdles to be overcome in achieving this.

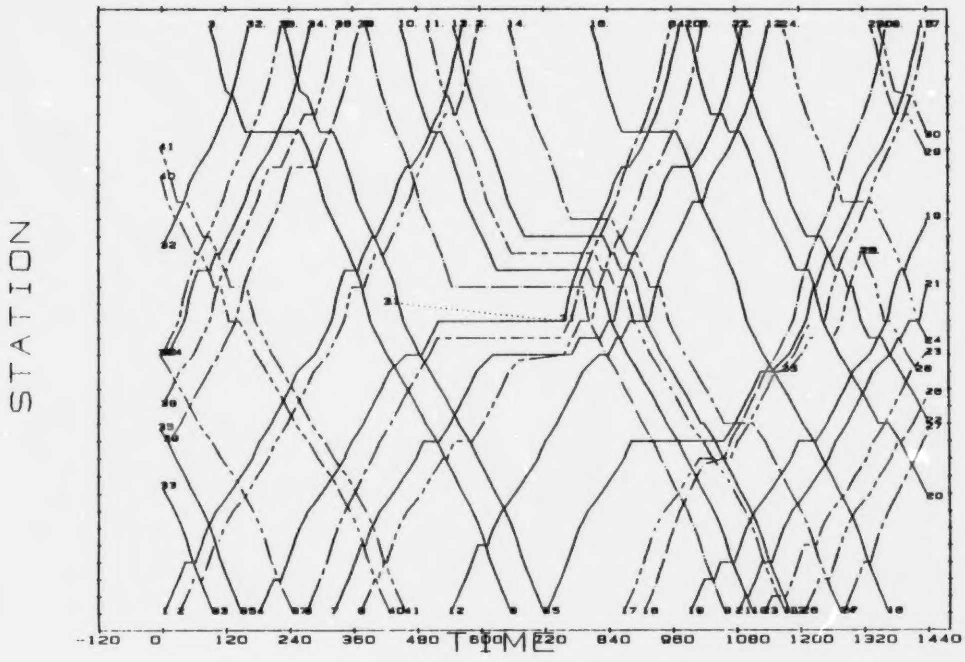
5.3 Value of Enhanced Power

While additional power is of no value in bulk applications without added tractive effort, it is of direct benefit in express service.

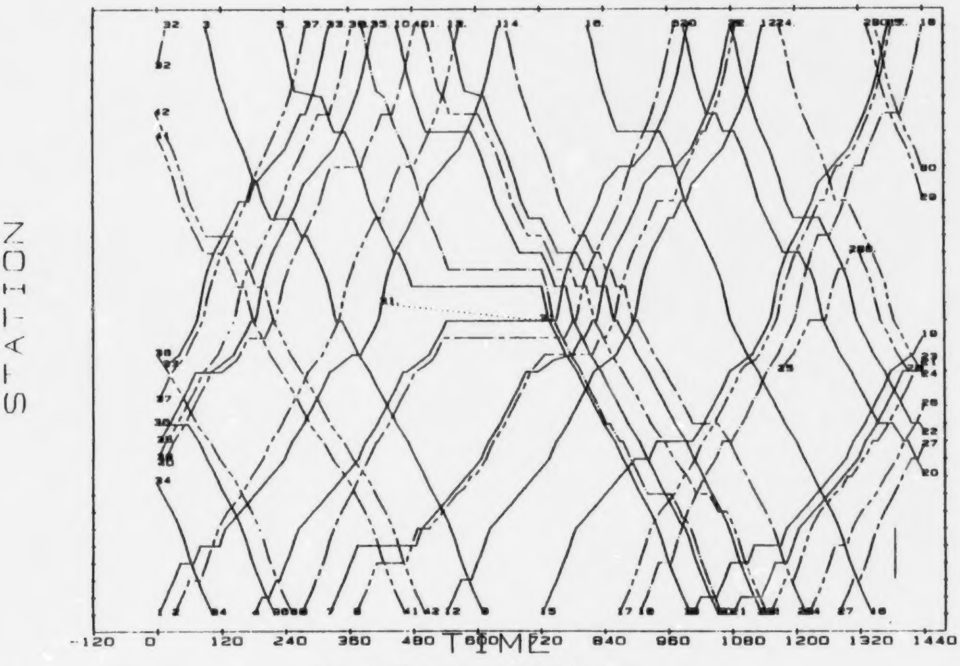
Express trains are the class of trains that carry time-sensitive and truck-competitive traffic. Locomotives are dispatched on a power basis, with the law of diminishing returns dictating power-to-weight dispatch ratios of 1.6 to 2.4 hp/ton. The diminishing benefits of added power beyond this range were investigated previously, as illustrated in Figures 5.7 and 5.8.[36]

The benefits of additional power under the assumption of no adverse impacts on operating unit costs is illustrated in Table 5.2.

The table indicates the value of reducing the number of axles per unit as well as increasing the power per unit. In comparison to the bulk case where enhanced tractive effort without corresponding power increase led to lower speeds and line capacity considerations, there are minimal negative impacts in the express situation. Additional power without tractive effort enhancement only affects the acceleration of the train at very low speeds. The primary concerns in the express situation are reliability, flexibility and utilization impacts.

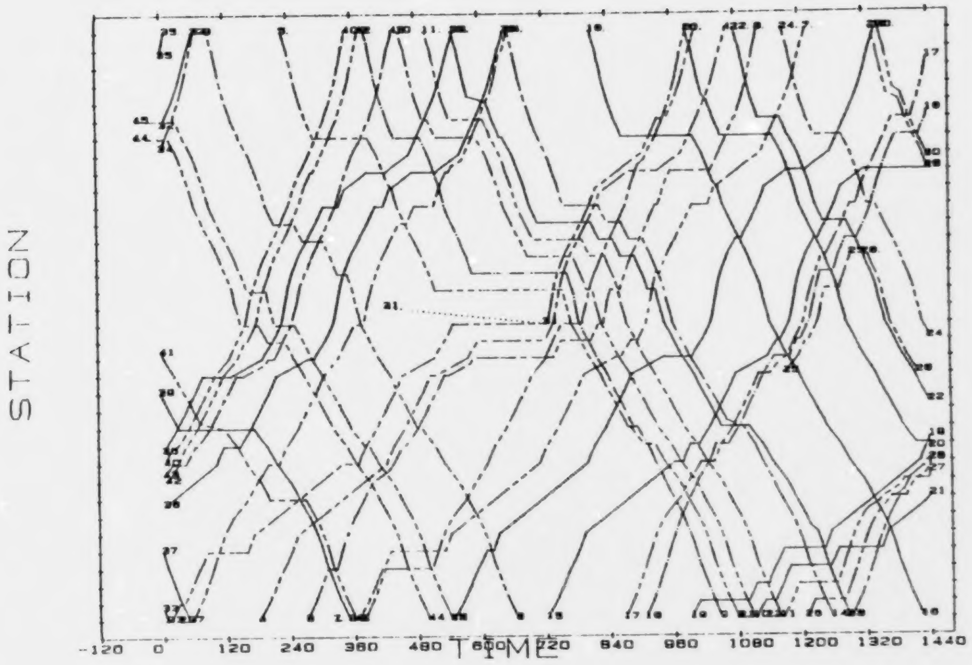


(a) Bulk train at 1 hp/ton



(b) Bulk train at 0.66 hp/ton

Figure 5.6 Train time distance plots



(c) Bulk train at 0.54 hp/ton

Figure 5.6 Train time distance plots (cont'd)

TABLE 5.2

INDIFFERENCE PRICES FOR DIFFERENT POWER LEVELS
(M$, after tax)

Power (hp)	3000	3000	3800	4600	6000	6000
Axles	6	4	4	4	4	6
Buy New	1.7	2.2	3.21	4.21	5.35	4.88
Rebuilt	.8	1.48	2.30	3.09	3.98	3.54

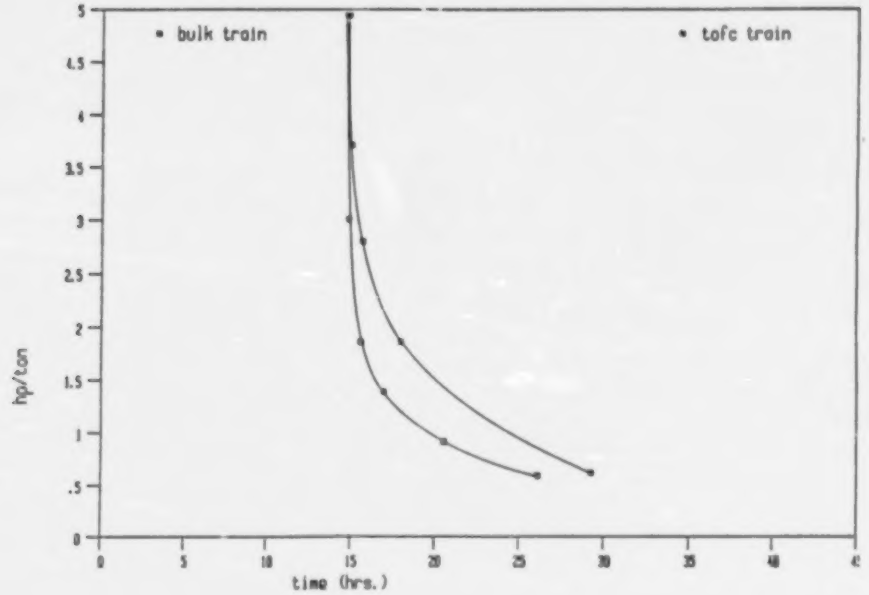


Figure 5.7 Horsepower per ton versus segment transit time, 794-mile, partial-double-tracked prairie segment

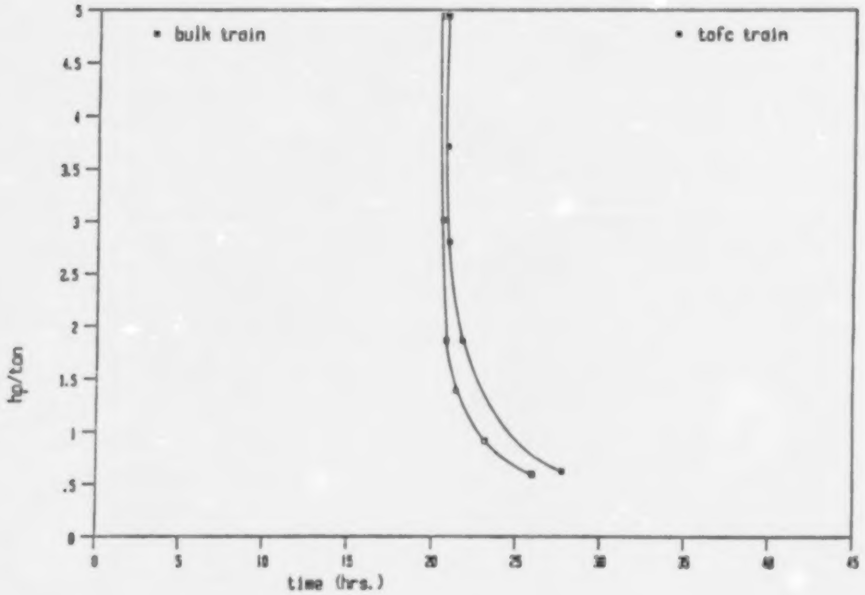


Figure 5.8 Horsepower per ton versus segment transit time, 748-mile mountain segment (low gradients)

Reliability is a concern because of the its value to the shipper for the types of commodities involved. At 12.3 road failures per year, a locomotive has a failure probability of 0.1 percent per trip. With multiple units the train can still complete its journey but with some delay. As the number of units is reduced, the overall probability of failure in the consist is reduced but the consequences of failure are more severe.

5.4 Cost of Specialization (or Value of Universality)

As noted in Section 2, most express trains operate to a fixed schedule, often filling out the train with decreasing priority traffic. Blocks of traffic are dropped off and picked up at major intermediate points (averaging a 1,000-mile stage length). Few unit trains are operated and motive power is drawn from the general service pool. Thus, flexibility and utilization influences are much more important in this application.

Using representative average numbers, a 2 hp/ton minimum dispatch ratio and a train length constraint of 100 cars at an average of 55 tons/car would lead to an allocation of four base-case 3,000 hp locomotives. These four locomotives could be replaced by two 6,000 hp locomotives or one 12,000 hp locomotive. However, the flexibility of these larger power blocks is reduced in response to variations in train weights. If an average car weight of 40 tons is encountered, three base-case locomotives are required, while the alternatives are unchanged. Thus, the in-service productivity improvement will not be a simple ratio of unit power. A flexibility derating factor is required. Also, utilization will decrease with specialization. A unit, once released from service and available for reassignment, will have a longer wait if it can only be assigned to a specific type of train rather than the next train.

These influences were evaluated by computer simulation of the movement of locomotives in terminal operations. The data used in the analysis were those reported by Mao & Martland[37] from their time study of motive power movements in a major U.S. railway yard. Other assumptions and the analysis are reported in more detail in Appendix G. The results of the productivity versus unit power assessment are summarized below in Table 5.3.

TABLE 5.3

DELAY SENSITIVITY TO UNIT SIZE

Percent 6,000 hp Units	0.00	24.75	100.00
Total Train Delay	100	126.2	147.3
Total Horsepower Dispatched	100	100.7	108.5
Total Average Locomotive Delay	100	178	168

These results provide only an indication of the types of affects one would expect with higher power sizes. The actual experience may vary significantly from this simulation for different yard activity levels and dispatch strategies. To a certain extent, there are tradeoffs between locomotive standing time and train delays awaiting motive power. The base-case statistics do not reflect actual Canadian operations. Still, the relative sensitivities can be expected to be reasonable estimates for a medium-sized yard's activity. Each of the three statistics results in a reduced replacement ratio for higher-powered locomotives. The train delay is largely derived from an increase in the number of cars left to wait for the next train rather than dispatching an additional locomotive at a low utilization. This number may be reduced by increasing the number of locomotives and allowing trains to be dispatched at higher power/weight ratios. The remaining two statistics present the main direct impact on the number of units required. The horsepower dispatched illustrates that 8.5 percent more horsepower is dispatched if only 6,000 hp units are used in place of 3,000 hp units (i.e. excess power is dispatched). The locomotive terminal queuing delays also increase with a reduction in the number of units. The 5 percent of the base-case locomotives service cycle in yard service pools would increase to 8.4 percent for the "all 6,000 hp case." The combined effect for this simulation would result in a replacement ratio of .5*1.085/(1-0.034) or .562 rather than .5 six thousand hp locomotives for each 3,000 hp locomotive.

Reductions in utilization versus degree of specialization were also assessed. These results are summarized in Table 5.4. The train and locomotive delays are further amplified by special-service locomotives. In this case, much of the delay is experienced by the general pool 3,000 hp locomotives which are given a lower priority in allocation to bulk and express trains.

The effect of a lower utilization on the capital purchase value was assessed for each of the high-power units presented previously in Table 5.2. In each case the reduced purchase price could be approximated as a multiple of the relative utilization (i.e. at .9 of the base case utilization the purchase value was .9 of the value shown in Table 5.2.

TABLE 5.4

DELAY SENSITIVITY TO DEDICATED SERVICE

	Base Case	Dedicated Bulk	Dedicated Bulk & Express
Total Train Delay	100	160.0	265.3
Total Horsepower Dispatched	100	81.9	82.2
Total Average Locomotive Delay	100	196	282

6. ACHIEVING HIGH TRACTIVE EFFORT OPERATIONS

The achievement of high tractive effort by a locomotive is influenced by many factors. For a desired dispatch adhesion level (defined here as total tractive effort available at the rail head/loaded locomotive weight), all of the factors shown in Table 6.1 must be taken into consideration.

TABLE 6.1

MITIGATING FACTORS TO HIGH ADHESION DISPATCH

- Available friction
- Normal force reduction:
  (a) fuel consumption
  (b) vertical dynamics
- Load-sharing characteristics:
  (a) weight transfer
  (b) wheel diameter
- Curve negotiation
- Traction motor reliability
- Torque transmission

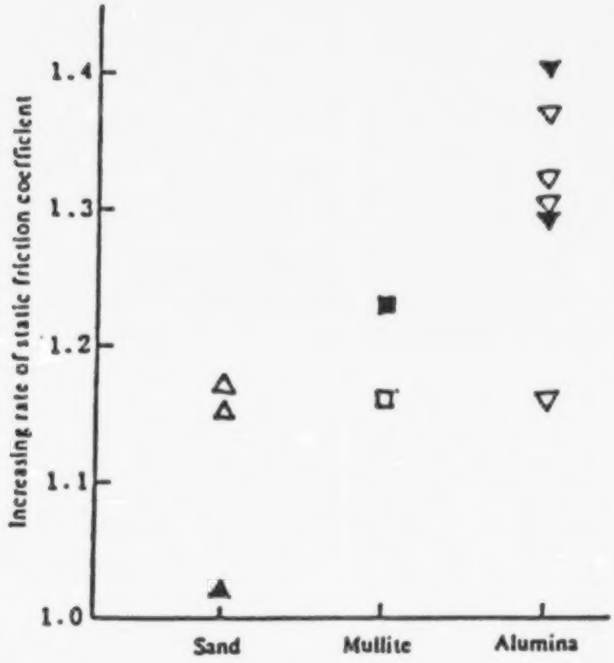
6.1 Available Friction

First and foremost is the factor of available friction between wheel and rail. The theory of friction-creep laws for rolling contact has been rigorously developed by Kalker for conditions of dry friction[38] and widely accepted and applied in wear and curving analysis. However, for locomotive adhesion dispatch, wet or contaminated rail is the design condition of interest. The theory in this area is less well developed. The Japanese National Railways (JNR) have recently undertaken an experimental investigation of adhesion under water lubrication.[39] The experiments show a lower initial slope of the traction coefficient (defined here as tractive effort at the wheel tread divided by wheel tread normal force) versus creep curve and a decreasing value with higher creep values. The shape is similar to those friction-creep curves published by Electro-Mechanical Division (EMD) of General Motors[40] and the Deutsche Bundesbahn (BD)[41] on the basis of field testing of locomotive performance.

In terms of enhancing available friction, the railway standard world-wide is the application of sand. During the late 1960s and early 1970s various alternatives to sand were investigated in several countries.[42] Many were found to be superior friction-enhancing agents but none was found to be technically and economically superior in the rail application.

One alternative investigated used trackside applicators of chemical solvent similar to existing gauge face lubricators. This system was considered to have many drawbacks in comparison to locomotive-based sand substitutes. The chemical is slow acting and must be applied continuously rather than with the passage of trains. Also, the increased friction of the rail head would lead to higher flange forces on curves for every vehicle in the train.

The most readily implemented of the locomotive-based alternatives was alumina, which demonstrated a 15 percent improvement over sand in tests performed by the JNR.[43] As with sand, it need only be applied when necessary. Thus, consumption is lower, and the adverse effects of high flange forces and wear rates on curves are reduced. Figure 6.1 presents the results of the JNR tests, while Table 6.2 compares the properties of the materials tested.



Source: Ohno Kaoru et al., "Low Adhesion Between Wheel and Rail Caused by Fallen Leaves and Its Remedy," *Quarterly Reports*, Railway Technical Research Institute, Vol. 24, No. 3.

Figure 6.1 Comparison of increasing rate of static friction coefficient from JNR tests

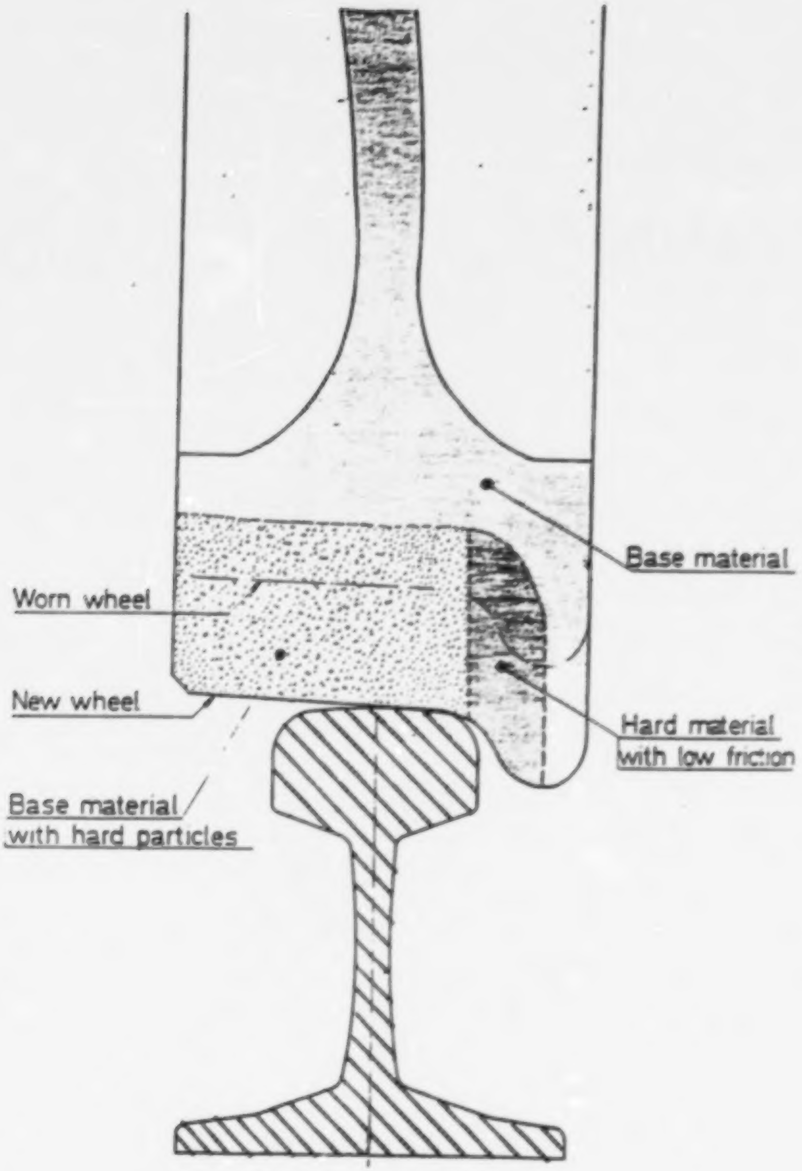
TABLE 6.2

COMPARISON OF MATERIALS TO INCREASE ADHESION COEFFICIENT

	Sand	Mullite	Alumina
Chemical formula	α-$SiO_2NaAlSi3O_8$	$3Al_2O_3 \cdot 2SiO_2$	α-Sl_2O_3
Mohrs' hardness	6.5	7	9
Angle of repose	31	35	37
Apparent density	1.4	1.4	2.0

For all of CN's system and all but the mountain region of CP, the requirement for peak adhesion is short term. In these instances the short-term overload capacity of the traction motor can be used if the friction is available at the wheel-rail interface. Thus, the 15 percent improvement may be realized with only an investment in the sand substitute. The value of a 13 percent improvement in adhesion was shown in Table 5.1 to yield an incremental value of $900,000 per locomotive. Whether used in existing locomotives or in new ones, this is a significant return for a low cost item.

Alternative wheel treads have also been proposed. Nitonol underwent a number of laboratory investigations because of its high friction performance.[44] Its friction creep characteristic was found to be very unstable, producing sudden large swings in friction. A new wheel based on the principles of powder metallurgy has been developed by ASEA Inc. Based on the fact that different parts of the wheel are subjected to different loads and are utilized for different purposes, a wheel having different properties along the tread and flange profile was fabricated. The powder used for the tread is mixed with particles of hard material which provide good adhesion. The inside of the wheel flange is heat-treatable and becomes harder than the rest of the wheel. Also, for wear reduction powder giving a lubricating effect may be added. A roughened tread surface should lead to improved ratio of material to water film contact, reducing the effects of elastohyrodynamic lubrication in the presence of contaminants. Ohyama[45] has shown the effectiveness of rough surfaces in realizing higher friction in the presence of water. The hard tread surface would lead to reduced wheel wear and higher rail wear. Unlike sand or alumina, the high friction is there whether it is required or not. Thus, flange forces of the locomotive would be higher on all curves. Although this is offset by the use of a low friction flange and the fact that the friction is not increased for following cars. Without any test information on the performance of the wheel it is difficult to quantify the economics of such a wheel. As with alumina, it warrants further investigation.



Base material
Worn wheel
New wheel
Hard material with low friction
Base material with hard particles

Source: ASEA

Figure 6.2 ASEA wheel

6.2 Normal Force Reduction

Given the friction available between wheel and rail, the next parameter of the equation is the normal force which exists at the boundary. There are two mitigating factors here -- the long-term effect of fuel consumption and the short-term effect of vertical dynamics of the axle. Fuel consumption takes on added importance with higher unit power ratings. With the existing range objectives requiring six liters of fuel capacity per tractive kW, the efficiency of the tractive coefficient decreases with power at the rate of -2.9 percent/MW.

Vertical dynamics is more of a concern for high-speed operations, but can still influence the design of the gear drive. Figure 6.3 illustrates the simulated effect of track quality on vertical unloading of a six-axle locomotive with axle-mounted versus frame-mounted traction motors. The time duration of these unloadings on mainline quality track is not sufficient to permit significant axle slip. However, the large swings in reaction torque when operating at the limits of available friction can impose large dynamic torsional loads on the mechanical transmission. This is a much accentuated problem for frame-mounted traction motors where the added elasticities on the drive can lead to resonant conditions with extremely high dynamic increments. From the viewpoint of adhesion performance at low speed, the benefits of reduced vertical dynamics associated with frame-mounted traction motors may well be overshadowed by the exacerbated torsional resonance problem.

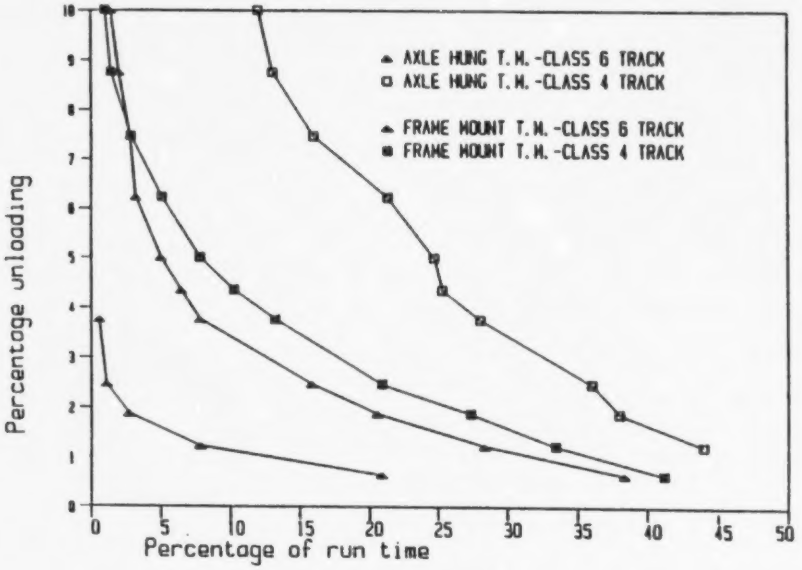


Figure 6.3 Axle unloading due to vertical dynamics

6.3 Load-sharing Performance

Table 6.1 presents two factors under the heading of "Load Sharing." This term is used to denote the ability of the propulsion system to distribute load in cases of unbalanced tractive forces and/or differential speeds. From the locomotive design viewpoint, it is a dual concern involving the marriage of mechanical design, which can be used to limit the imbalance, with the electrical design which must accept the imbalance. Unfortunately, the operating environment, as will be shown below, places conflicting demands on performance. The determination of the most economical system is extremely complex.

The effects of weight transfer on tangent track adhesion performance is illustrated in Figure 6.4 for several propulsion characteristics.

The weight transfer is that of the base-case locomotive, which achieves a ±7 percent load differential between the lead and trailing axles of the locomotive. The dc traction motor torque-speed curves are based on a computer simulation using laboratory test data collected by Garret[46] for the D77 traction motor used in the SD40, and a shunt version of the motor which Garret designed. The slip characteristic of the ac traction motor is based on a 0.6 Hz slip at rated torque and a variable voltage variable frequency (VVVF) inverter which has a 160 Hz frequency at 70 mph top speed. Each traction motor curve is based on a 20 percent adhesion at the trailing axle (i.e. the upper friction curve) at 11 mph ground speed.

The limitations of the dc traction motor in a common bus connection are obvious. The lead axle with a series-connected motor must be able to creep at 6 percent in order to accommodate the 15 percent difference in tractive effort. If there is a falling friction slip curve, then the power to all traction motors must be reduced to get an intersection of friction and motor torque on the lead axle. As higher tractive efforts are sought, the separation of the upper and lower saturation plateaux increases in nominal terms. The ac traction motor, on the other hand, accepts this weight transfer effect with no concern of motor run away. The concern of differential current loading still remains, however. The current imbalance is proportional to the torque imbalance in each case. Thus, the trailing axle will have a 7 percent overload when operating at the locomotive nominal rating.

Figure 6.5 presents a condition of tread wear on the lead axle of the locomotive. The lower curve represents a 5 mm differential wear of a nominal 1,016 mm diameter wheel. For unit train service where the lead axles in the loaded direction are the same as ones for long intervals, tread wear can be expected to be worse for the series dc case because of its higher creep rates. However, as the figure illustrates, differential wear is more of a concern for the steep torque-slip characteristic of ac traction motors.

For the ac traction motor there is a 33 percent difference in current loading for the lead and trailing traction motors.

Figure 6.5 illustrates that differential tread wear is not a direct limitation to adhesion under low friction conditions but forces higher creep rates and higher differential motor loadings to yield the same tractive effort.

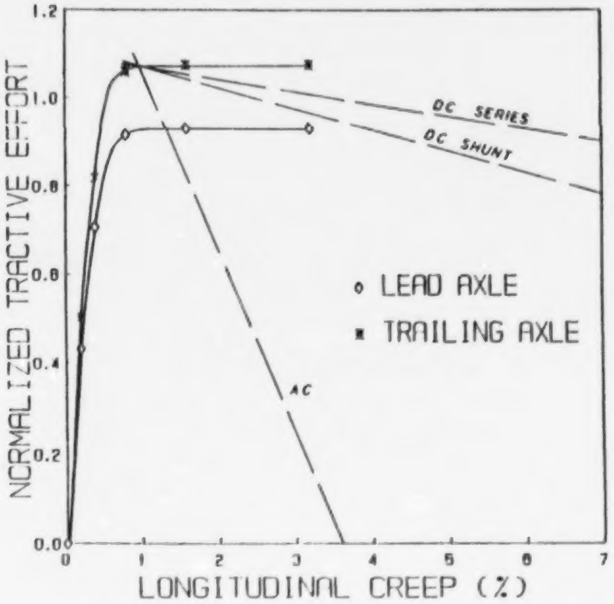


Figure 6.4 Load-sharing characteristics

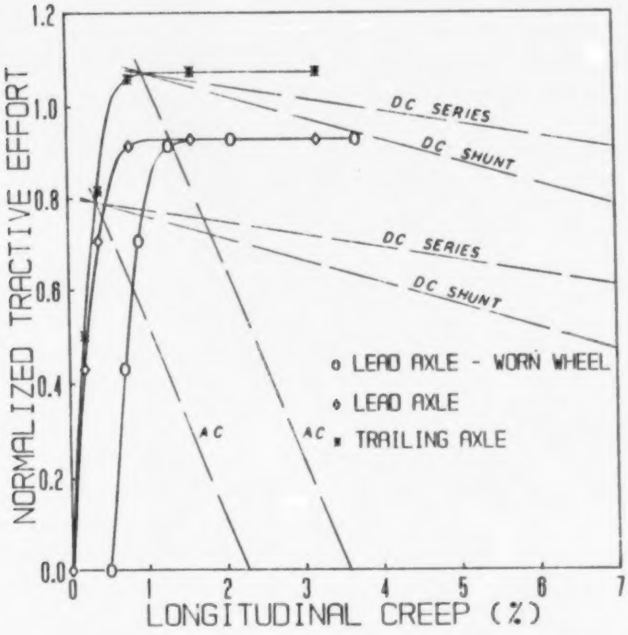


Figure 6.5 Worn wheel load sharing

It means that the wheels must be allowed to creep at a sufficient rate to enter the saturation region. Under high friction conditions, the adhesion performance is limited by the current limit of the traction motor. At the 5 mm differential wear, the traction motors would have to have current ratings of 15 percent above the nominal traction rating.

This concern is exacerbated with increasing power levels. If this same tractive effort was to be provided at 22 mph (i.e. twice the power), the steepness of the torque-slip curve would double and the traction motor would require a 50 percent rating above nominal. Without this over design, the current limit would restrict the adhesion performance of the locomotive. Thus, the ac traction motor is faced with the paradox of being able to achieve a higher tractive effort under lower friction conditions than it does under higher friction conditions.

These concerns are, of course, removed if wheel diameters are maintained to a close tolerance. For the wheels on the locomotive, any differences are self-correcting due to the higher tractive force exerted by larger wheels which leads to a faster wear rate. However, it does mean that when a truck or a wheel is changed out, its replacement must be within the allowed tolerance in relation to the other wheels on the locomotive, or both trucks need to be changed out with pre-matched wheel sets. In either case, some incremental maintenance costs can be expected.

The benefits of a steep torque/speed curve on tangent track are obvious. However, the advantages of a steep torque-slip characteristic are lost in situations of curve negotiation.

6.4 Curve Negotiation

For most railways, the steep gradient conditions of mountainous terrain are accompanied by tight radius curves. This is also true for Canadian railways, although the conditions for BCR and CP Rail are more severe than for CN Rail. On CN's mountain line, peak adhesion must be maintained on up to 4 degree (436 m) curves, while for CP's parallel route peak adhesion is required frequently on curves up to 8 degrees (218 m).

The effect of curvature on the traction coefficient can be severe. The curving characteristics of a conventional North American three-axle locomotive truck are such that lateral creepage exists at the wheel/rail interface of the lead axle. The implications of curve negotiation for high-adhesion locomotive design were investigated by means of computer simulation with a dynamic curving model developed by Fortin.[47] The model and assumptions are discussed in Appendix E.

For the lead axle, the influence of lateral and spin creep components at the wheel/rail interface is to reduce the friction available for longitudinal traction. When operating in the saturation zone of the friction-creep curve, with a contact patch aspect ratio typical of wheel tread on rail head, Kalker's friction creep theory can be approximated as a vector sum of longitudinal and lateral creep rates. Thus,

$$F_x = \frac{f_{11}\, e_x}{[\, (f_{11}\, e_x)^2 + (f_{22}\, e_y + f_{23}\, e_s)^2\,]^{1/2}}\ \mu N$$

where: e's are creep rates
f's are creep coefficients
μ = coefficient of friction
N = normal force
F_x = longitudinal force.

It is evident that the available friction can be utilized for longitudinal traction only if the longitudinal creep is allowed to increase substantially beyond the lateral creep rate. However, a low longitudinal creep rate is sufficient for the non-flanging trailing axle.

The predicted force-creep curves of the lead and trailing axles of a conventional three-axle truck for several curve radii are illustrated in Figure 6.7. Superimposed on the friction-creep curves are the torque-speed characteristic curves of three traction motor types.

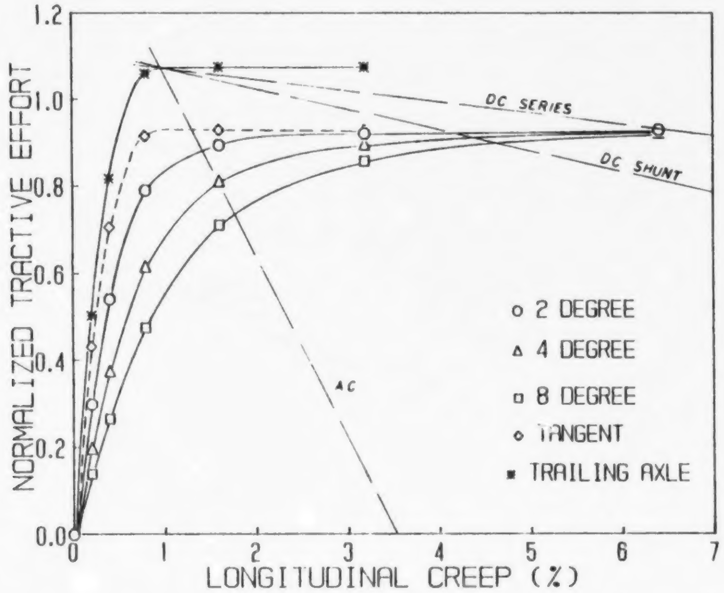


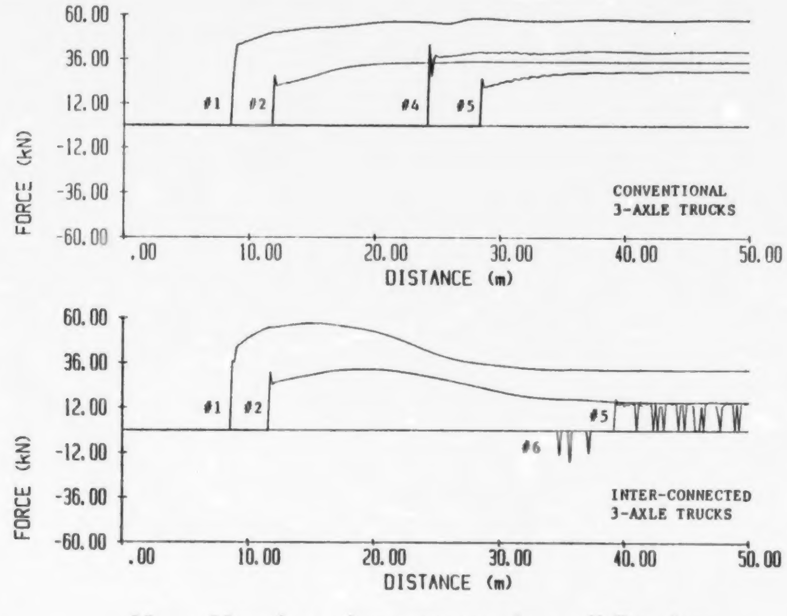
Figure 6.6 Friction creep characteristics in locomotive curving

For dc traction motors, curve negotiation presents no added concern. However, the ac traction motor's steep torque-slip curve is a detriment to achieving high tractive effort in curve negotiations. At low friction levels is must permit high creep rates on all axles to get maximum utilization of available friction. In high friction situations it must accommodate large current load differentials. In the curving situation this can be a short-term overload rather than continuous rating.

These load-sharing disadvantages of both ac and dc traction motors can be overcome by going to individual axle control, or may be mitigated by mechanical design.

Table 6.3 shows the interaction of mechanical design alternatives and propulsion alternatives for tangent and curved track. The table is based on the operating characteristics of Figure 6.6 in achieving the maximum tractive effort possible. It can be seen that for high adhesion, performance in curve negotiation mitigates the benefits of reduced weight transfer design and a steep torque-slip curve. A reduction of the weight transfer coefficient (defined here as percent unloading/ percent adhesion) from 0.3 to 0.05 reduces average full-friction-utilization wheel slip from 4 percent to 1.6 percent for a series dc drive on tangent track, whereas an increase from 4 percent to 6.2 percent occurs on an 8 degree curve. Similarly, while average full-friction-utilization wheel slip is reduced significantly with ac traction on tangent track, it is higher than the series dc case on an 8 degree curve. Individual axle control is seen to be much more effective than reduced weight transfer.

The benefits of adding a simple inter-truck connector to conventional three-axle trucks was investigated. A reduced angle of attack is achieved which reduces the lateral creep rate. The simulated curving performance is compared, in terms of flanges forces, in Figure 6.7. In terms of lateral creep, the device produces values typical of a curve of two times the radius (or one-half the degree of curvature). This is a significant benefit for ac traction, particularly on curves up to 4 degrees, as one can see from Figure 6.7. If a radial orientation could be achieved with a full steering truck then the concerns of curve negotiation for asynchronous ac traction would be eliminated.



Note: Negative values represent low rail flanging

Figure 6.7 Flange force comparison in a 4 degree curve

TABLE 6.3

TRACTION EFFICIENCY AND AVERAGE AXLE
WHEEL SLIP AT MAXIMUM FRICTION UTILIZATION

Weight Transfer Effect (percent unloading/ percent adhesion)	Common Bus Series dc	Common Bus ac	Individual Axle Control
Case A - Tangent track (26 percent friction)			
0.30	93 (4.0)	93 (1.6)	93 (1.5)
0.15	96 (2.9)	96 (1.6)	96 (1.5)
0.05	98 (1.6)	98 (1.5)	98 (1.5)
Case B - Tangent track (42 percent friction)			
0.30	89.5(5.7)	89.5(1.8)	89.5(1.5)
0.15	94 (3.6)	94 (1.6)	94 (1.5)
0.05	98 (2.2)	98 (1.5)	98 (1.5)
Case C - 8 degree curve (26 percent friction)			
0.30	93 (4.0)	93 (6.9)	93 (3.8)
0.15	96 (5.5)	96 (6.9)	96 (3.8)
0.05	98 (6.2)	98 (7.0)	98 (3.8)

6.5 Reliability

Reliability is an important factor in evaluating alternative methods of achieving higher dispatch adhesions. For present operations, the dispatch level reflects traction motor reliability as well as wheel-rail traction coefficients. There are two factors to consider -- the increasing probability of failure with increasing torque levels and the implications of individual failure for total consist tractive effort. Present Canadian experience is a mean time between failure (MTBF) for traction motors of about 30 months. This may be higher than the North American average because of snow ingestion problems in winter (50 percent failure in 24 months is the indicated U.S. experience[48]). An eight-month, non-winter sample of traction motor failure causes showed 37 percent to be related to armature failures and 11 percent to pinion failures. Each of these failure modes will be exacerbated by higher adhesion operation. With existing performance, a five-locomotive consist has a 6.25 percent chance of experiencing a traction motor failure in a two-day coal-haul service. A single failure for the existing locomotives with a series connection of traction motors means a 7.1 percent increase in friction is necessary. Table 6.4 illustrates the implications of several configurations at assumed reliability. The impact factor in the final column is the product of the probability of failure and the friction enhancement which failure necessitates. The product is normalized with the existing situation as represented in the first row set to 100. The benefits of a parallel connection of traction motors is evident. It can also be noted that a reduction in traction motors can result in a reduced impact factor, even if the MTBF is decreased as a consequence of higher tractive effort per axle.

TABLE 6.4

RELIABILITY FACTORS IN ADHESION

Enhanced Fiction					
Assumed MTBF$ (months)	Number of Powered Axles	Series or Parallel Connection	(a) Probability of Failure (percent)	(b) Extra Adhesion Required Given a Single Failure (% of original)	(a)*(b) Impact Factor
31	30	Series	6.25	7.1	100
31	30	Parallel	6.25	3.5	49
20	20	Series	6.4	11.0	159
20	20	Parallel	6.4	5.3	76
60	20	Series	2.2	11.0	54
60	20	Parallel	2.5	5.3	26

6.6 Transmission Systems

The final step in realizing higher tractive efforts is to provide the increased torque corresponding to the higher adhesion levels, or alternatively to increase the number of axles per locomotive.

Increased Torque. As noted previously, the tractive effort characteristic for a given traction motor can be scaled by the gear ratio used. Existing dc traction motors realized a range with 24 percent adhesion at low speed and 73 mph to speed with 42 inch wheels and 70:17 gearing. Provision of this range requires the traction motor to be oversized. It must be signed to take high currents at low speed. Because its field-weakening range is limited to about 60 percent, it must achieve higher speeds by increasing supply voltage. Thus, it must be designed for high current and high voltage, but neither are applied concurrently.

With the existing limits of current and voltage, increased tractive effort only comes with decreased top speed and there are limitations to the size of gears which can be used. The implications have been discussed by Corneil et al.[49]

A high gear ratio requires a relatively small gear, with few teeth, on the traction motor shaft, and a relatively large gear on the locomotive axle. The traction wheel diameter limits the diameter of the gear mounted on the axle. Clearance between the rail head and the gear housing is essential if the locomotive is to traverse switches and cross-overs without damaging the gear housing. Additional clearance is required if rocks and other obstructions which might fall on the track are to be cleared by locomotive structural elements other than the gear housing.

The outside diameter of a standard cut gear is (N+2)/P, where N is the number of teeth in the gear, and P is the diametral pitch (the ratio of the number of teeth to the number of inches in the pitch diameter). Thus, a 77-tooth gear having a 2.25 diametral pitch will have an outside diameter of 35.11 inches. Allowing 5/16 inch for gear clearance and enclosure thickness, the diameter of the gear housing becomes 35.736 inches. A fully worn 42-inch wheel (39-inch diameter) provides a clearance between the gear case and the rail head of only 1.632 inches. Anything less than this would not be considered adequate.

The pinion gear, of smaller diameter, will have a relatively small number of teeth. Because each tooth will engage more frequently to carry the load, the wear and fatigue criteria for the pinion gear are more critical. Reducing the pinion to 15 teeth would reduce the root diameter to 5.638 inches, and would result in additional difficulties in mounting the pinion on the motor shaft. In addition, a 15-tooth pinion would have a wear life (in kilometers) only 48 percent of that of a 17-tooth pinion.

An increase in gear quality can increase gear life. However, increased accuracy and improved surface finish would substantially increase the cost of gear production. Also, it is necessary to install roller bearings on both the axle and the traction motor to reduce the bearing clearances, to more rigidly mount the bearing so as to maintain centre-to-centre distances, and to increase the stiffness of both the axle and the traction motor shaft to reduced deflection.

Thus, higher tractive efforts can be achieved with the existing traction motor, but only at the expense of top speed and pinion cost and life. The estimated performance is shown in Table 6.5.

TABLE 6.5

ESTIMATED COST AND LIFE PERFORMANCE OF PINION GEARS

	Gear Ratio		
Gear ratio	70:17	77:17	77:15
Nominal adhesion (%) (42-inch wheels)	24	26	30
Top speed (mph) (with worn wheels)	68	62	54
Normalized pinion life	1	1	.48

These values are based on an SD60 locomotive with A D87 traction motor. Canadian General Electric (CGE) and General Electric (GE) traction motors are larger and have a higher torque rating. Thus, the level of adhesion for the same gear ratio could be increased by 10 to 20 percent if a retrofit were made. Expanding performance beyond these levels requires development of a larger dc traction motor and a new truck to accommodate it, or different technology. ASEA and Brown Boveri Corp. (BBC) both build 1,300 hp dc traction motors for use in Europe. However, they are too large in diameter to fit in a North American locomotive truck with 42-inch diameter wheels. They are used in locomotives with 48- to 54-inch wheel diameters. Larger wheel diameters have several repercussions, the most significant being weight. The additional unsprung mass of larger-diameter wheels is a particular concern at high vehicle speeds. Traction motors on European locomotives are frame-mounted, with flexible coupling to the axle in order to reduce the unsprung mass. Transfer of this practice to North American locomotives is inhibited by higher torque transmission requirements in combination with an industry requirement of ±2.5 inches of axle freedom so the suspension does not bottom out on rough track. The other concern is for locomotive height -- both for clearance and centre of gravity. Thus, while it is feasible to achieve much higher power transmission capacities with dc traction, considerable development would be required. Again, while many of the industry-imposed constraints could be relaxed for a specific limited-service application, this would reduce its flexibility.

6.7 Rail Wear and Force Effects of High Adhesion

Flange Forces in Curves. An increase in tractive effort in curve negotiations is accompanied by a decrease in lateral tread forces. The simulation results for a conventional three-axle truck at increasing tractive effort are shown in Figure 6.8. The flange force of the lead axle is shown for conditions of 40 per cent friction. An increase of adhesion from 24 to 36 percent reduces the peak flange force by 25 per cent in a 200 m radius curve.

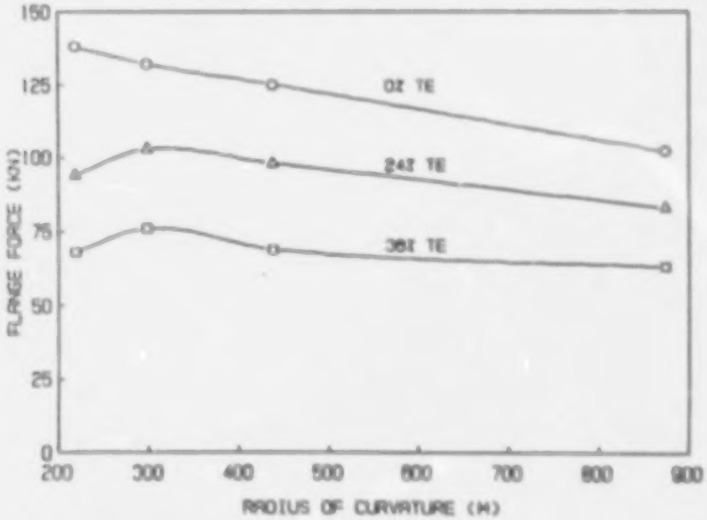


Figure 6.8 Lead axle flange force relation to adhesion

However, there is a net increase in wear rate of the rail gauge face due to the fourfold increase in wheel creep.

Higher traction-to-power ratios lead to lower speeds and thus under-balanced superelevation on curves. The curving performance was investigated for a 7 mph speed on an 8° curve with a 15 mph balance superelevation. A 350,000 lb drawbar force was applied to the trailing coupler. The full simulation results are part of Appendix F. The flange forces for both conventional and interconnected three-axle trucks are illustrated in Figures 6.9 and 6.10, respectively. While the lead axle angle of attack does not change considerably from the previous simulations, all flange forces are shifted to a lower position on the force axes. For the conventional truck, this leads to a 72 kN flange force on the low rail for the lead truck, an L/V of about 0.45. The interconbected truck again results in lower flange forces.

Increased Axles. The tractive effort of a locomotive can also be increased by increasing the number of axles. An eight-axle locomotive would give a 33 percent increase in tractive effort with the same traction technology. A longer platform or a reduced fuel capacity would be required to accommodate the extra axles on the locomotive.

Alternatively, one might install powered trucks on an adjacent load-carrying car as a means of increasing tractive effort without significantly altering the electrical propulsion capacity. The use of a series-parallel connection of traction motors, which has recently been displaced by higher-capacity alternators or alternator transition in North American locomotives, could be reinstated to achieved ten traction motors per locomotive with modest design effort. In unit train service the extra four traction motors on an adjacent car would need to be cut in for 15 percent of the time in the loaded direction only. The rest of the time they would be disconnected from the power supply bus. A conceptual design is presented in Appendix H. The additional weight of the powered trucks and the volume required for traction motor blowers and filters, reduces the payload capacity from 103.5 tons to 70 tons. As previously shown, the series connection of traction motors will reduce adhesion performance, via poorer load sharing and reliability. However, the reduction may be minimized by extra surveillance of tread wear on paired axles (1,4:2,5:3,6:7,9:8,10). The numerous stiff cables required would require some form of mechanical leverage to couple the car electrically. A different brush might also be necessary for the unloaded traction motors to prevent increased wear.

Wear. Wear and damage to wheel and rail are the main concerns of high adhesion operation. Table 6.6 presents a relative wear index on the basis of a wear proportional relationship of:

$$W \alpha (\tau_G)^{3.3} e$$

where: W = wear rate
τ_G = maximum octahedral shear stress
e = creep rate.

A 50 per cent increase in adhesion is seen to give a threefold increase in tread wear.

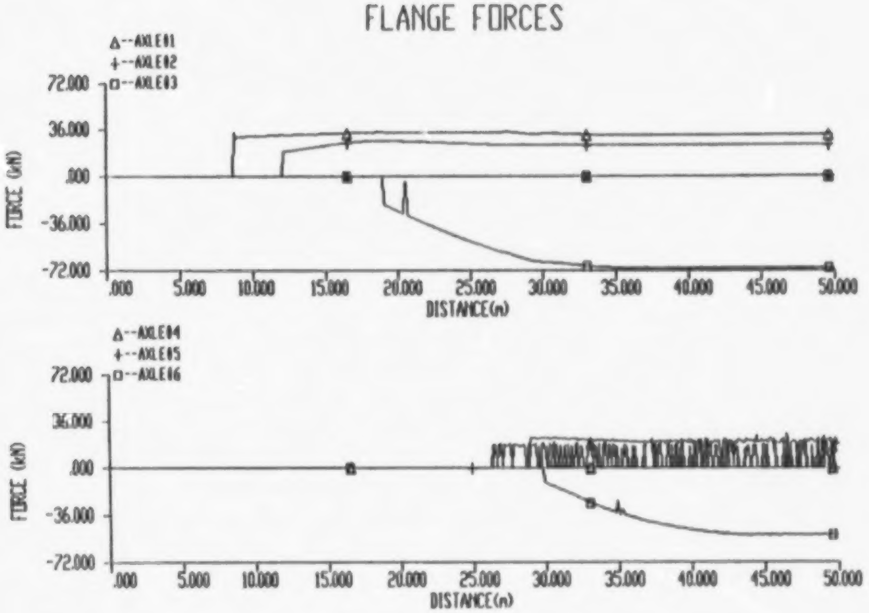


Figure 6.9 Conventional three-axle trucks under balanced curving performance (8° curve, 350,000 lb drawbar force, 24 percent adhesion)

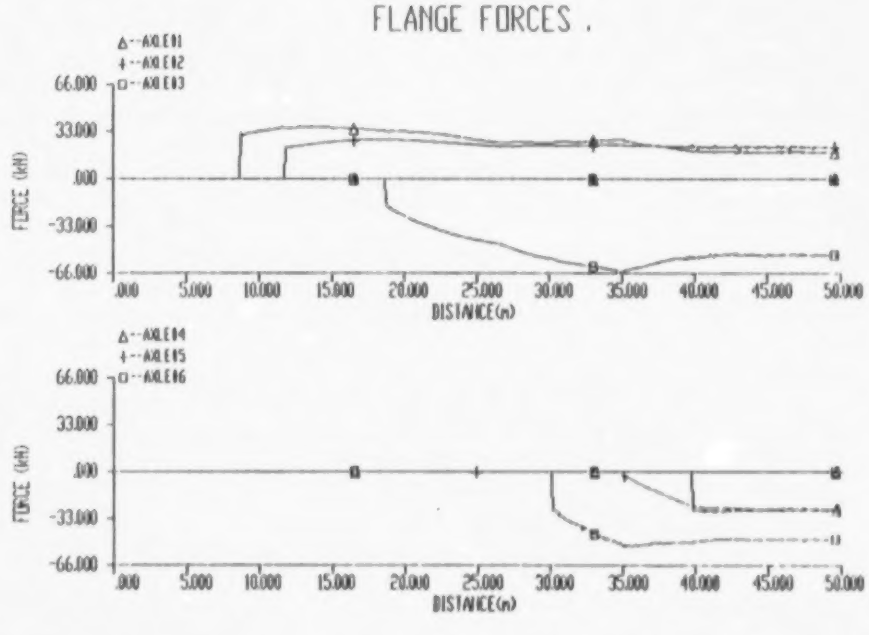


Figure 6.10 Interconnected three-axle trucks under balanced curving performance (8° curve, 350,000 lb drawbar force, 24 percent adhesion)

TABLE 6.6

TREAD WEAR INDEX ON TANGENT TRACK

Adhesion (%)	Friction (%)	40" Wheel	42" Wheel	54" Wheel
18	28	1.00		
24	28	2.45	2.21	1.36
24	40	1.47	1.32	0.81
32	40	3.94	3.55	2.19
36	40	4.37	3.94	2.43

High tractive efforts will also increase the incidence of plastic flow, corrugation, and surface fatigue, regardless of the creep rate. The main impact of increased wear will be on the wheel tread, whereas the impact of plastic flow and corrugation will be on grinding costs of low rails on curves. In each case the wear or damage index is known to increase at an exponential rate with increasing tangential force.[50,51]

The relative impacts of corrugation and plastic flow are presented in Table 6.7. The indices are based on a proportional relationship of:[52]

$$(\text{STRESS - SHAKEDOWN})^{7.8}$$

TABLE 6.7

INDEX OF LOW RAIL PLASTIC FLOW/CORRUGATION

			Index at Different Wheel Diameters		
Adhesion (%)	Curve Radius (m)	Friction (%)	40" Wheel	42" Wheel	54" Wheel
24	tangent	28	0.000	0.000	0.000
36	tangent	40	0.000	0.000	0.000
24	872	40	1.740	1.610	1.167
24	872	28	1.142	1.047	0.1732
36	872	40	2.033	1.888	1.393

The reference base is the low rail load conditions of a loaded 91-tonne car (wheel load of 141 kN) with a 200 mm tread radius at point of contact on a 872 m curve. This condition represents the threshold of corrugation and plastic flow problems.

The results indicate that high adhesion operation will not lead to corrugation problems on tangent track. In the case of both wear and corrugation, the impact of higher adhesions can be mitigated by use of larger wheel sizes.

High utilization of available friction will not always be required. The available friction is a statistical variable with a normal distribution. A train which is dispatched such that there is a high level of confidence that it can successfully climb the steepest grade, will very seldom be operating at the limits of friction. If the required tractive effort can be met at low utilization of the available friction (e.g. 0.8 of the saturation level), then large creep levels are not necessary. Also, the incidence of high tractive effort itself is very low. For the tractive effort duty cycle shown in Section 2 for a loaded unit train run over CP's mountain main line, the demand for tractive effort above 70 percent of maximum is required less than 15 percent of the time and not at all in the empty direction.

Thus the incidence of high creep rates will be relatively low. Also, offsetting these incremental wear rates per axle, which are relevant to a small percent of the route length, is a reduction in the number of locomotive axle loadings over 100 percent of the route. Also, as previous illustrated in Section 2, the relative costs of wheel maintenance and track maintenance due to locomotives are small components of its full life cycle cost.

Figure 6.11 illustrates the sensitivity of the savings to possible operations cost impacts. The base-case costs of wheel maintenance and rail grinding and replacement are escalated by the locomotive's adhesion performance ratio raised to an exponent of 1.5 and 2.0 in the first two cases. In the latter case, traction motor armature and pinion maintenance costs are also escalated. The savings are shown to be relatively insensitive to these factors.

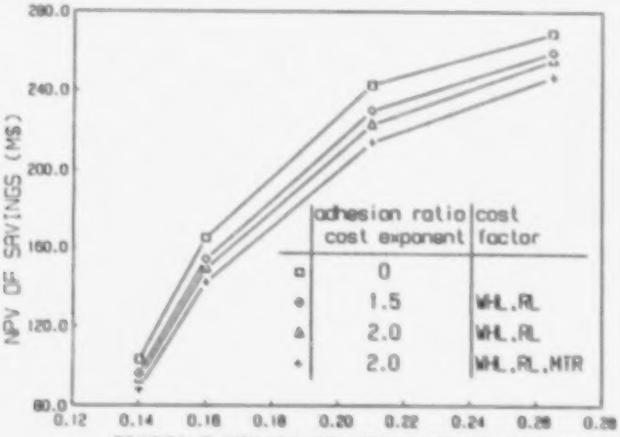


Figure 6.11 Sensitivity to increased wear costs

7. HIGH POWER ALTERNATIVES

The current-generation, four-axle, diesel-electric locomotive is at the limits of the technology employed. Power generation capacity is constrained by axle load limits imposed by the railways, while power transmission capacity is constrained by the space available in the existing truck design. With the same technology, one can realize more power per unit with additional axles. But for a four-axle unit, the constraints must be relaxed or different technologies utilized.

Weight constraints for the current-generation 3,800 hp locomotive were investigated. The increased weight for the variable components is presented in Table 7.1. It is not possible to meet the 65,000 lb axle load limit and at the same time provide the desired range capability associated with one Imperial gallon of fuel capacity per tractive-horsepower (thp) without substituting lighter-weight materials.

TABLE 7.1

WEIGHT ESTIMATES OF INCREASED POWER
FOUR-AXLE DIESEL-ELECTRIC LOCOMOTIVES
(based on conventional dc traction locomotive)

Components		Weight (lb)	
		Base-line	4,000 hp
Diesel Engine	ghp	3,000	4,000
	weight	32,300	39,500
Alternator	thp	2,750	3,700
	weight	13,000 (est.)	17,500
Fuel	imp. gals.	2,500	3,300
	weight	20,750	28,400
Fuel Tank		7,300	11,500
Supplies		4,800	6,400
Miscellaneous Add-on			3,000
Total Variable Weight		78,150	105,300
Increase			27,150
Projected Total Weight		260,000	287,150
Projected Axle load		65,000	71,787
Projected Overweight/axle		-	6,788

Those component weights most sensitive to increasing power are the engine, fuel (and tank), alternator and traction motors. Weight projections have been made for these components on the basis of the weight variations of existing systems in the range of 3,300 to 4,000 hp. The results are presented in Table 7.2. With the efficiency improvements achieved by current-generation diesels over the base case, it has been assumed that one U.S. gallon of fuel/ghp will provide a similar range capability to that achieved with 1 Imperial gallon/thp on the base-case locomotive.

TABLE 7.2

WEIGHT-TO-POWER RELATIONSHIP OF MAJOR COMPONENTS

Component	Weight-to-Power Relationship [W (lbs) = a + b * P (hp)] a	b
Diesel engine:		
Medium-speed, two-stroke	7,900	8.0
Medium-speed, four-stroke	8,100	9.62
Engine supplies	4,000	1.0
Fuel		7.0
Fuel tank (standard)	1,800	2.3
(improved UTS)	1,200	1.6
900 rpm. Alternator (@ .95 P)	1,000	4.0
1500 rpm. alternator (@ .95 P)	820	3.25
dc traction motor (@ .9 P / 4)	2,000	5.0

In relation to traction motor upgrading, the characteristics of ac traction motors are well suited to express train service requirements. Without the voltage and rotational speed limitations of the dc motor's commutator, much higher power densities can be achieved. In bulk-service operations, these advantages are mitigated by the high continuous current requirements imposed by a limited wheel diameter. In the case of CP Rail's prototype three-phase locomotive, current requirements associated with a 28 percent adhesion capability led to a motor with a rating of over 2,000 hp. In express service, the under-utilized voltage and rotational speed capacity of this motor could be more fully utilized (although the inverter must be sized to provide the extra voltage). Thus, with ac traction motors it is technically feasible to fit up to 2,000 hp per axle in an existing North American motor mount. For a locomotive dedicated to express service, it may be possible to reduce the continuous current rating of the motor and thereby its size and weight. The low-speed, high-current operating point of the motor in express train duty is only temporary and would be within the thermal overload capacity of the traction motor.

Unfortunately, each of the above developments adds weight to the locomotive, in addition to development and capital costs. Weight is the major constraint to power increases. The following alternative methods of overcoming weight constraints were evaluated for the base-case locomotive:

- weight reduction of frame and body components:
  (a) via design and materials changes
  (b) by relaxing buff load requirements
- increased number of axles per unit
- removal of fuel and tank to a separate tender
- use of a higher-speed, lighter-weight diesel engine
- use of a lightweight turbine engine
- use of a hybrid diesel/turbine power plant.

The technical evaluation of each alternative addressed not only its relative effectiveness in overcoming the imposed constraints, but also its disbenefits. The disbenefits range from front-end development cost only, to performance deterioration in other cost categories.

7.1 Weight Reduction

The firm of D.D. Taylor and Associates was retained to review weight reduction potential in existing four- and six-axle locomotives. Their assessment is attached as Appendix I. Table 7.3 summarizes the savings which can be realized from design and material changes. These changes are expected to involve only front-end capital cost increases.

Applying the weight variability factors of Table 7.2 and the estimated savings potential of Table 7.3, the limits of existing diesel-electric technology are 4,000 hp on four axles and 6,000 on six axles.

Fuel Tender. Removal of the fuel tank to a separate tender would realize a weight saving of 1,800 lb plus 8.6 lb/hp. Thus, from the variable weights of Table 7.2, the 4,000 hp limit of technology could be extended such that:

$$(8 + 1) * P + .95 * 4 * P + .9 * 5 * P = 1200 + (7 + 1.6) * 4000$$

where the solution for "P", the gross engine power increase, is 2,000 hp. Therefore by using a fuel tender, a 6,000 hp locomotive could meet the weight constraints with a medium-speed diesel engine.

TABLE 7.3

WEIGHT REMOVAL POTENTIAL

	Six-Axle Locomotive		Four-Axle Locomotive	
Item	Saving (lb)	Comment	Saving (lb)	Comment
Locomotive frame	30,000	EMD or BBD (based on previous study)	Baseline 2,500	From previous study. Four-axle beam "much closer to line."
Truck frame	5,800	Modified Dofasco	6,400	ZWT frame: 10,000 Centre post: 1,200 12,000 Est.new frame: 8,000 32,000
Fuel tank	4,000-hp 3,900 5,000-hp 4,000 6,000-hp 4,300	All using higher UTS steels to maintain puncture resistance with thinner walls.	3,000-hp 1,500 4,000-hp 3,900	
Structural above deck	7,500	Redesign	5,000 10,000	Redesign. Aluminum or very thin corrugated sheeting.
Ballast removal	3,000			
Further savings through relaxing the structural regulations.	--	Makes beam too flexible.	5,000	Estimate only, not calculated. Four-axle beam tends to be stiffer and some weight could be removed from beam itself without becoming too soft.

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* The effectiveness of reduced buff load requirements was limited by frame flexibility concerns. Diesel engines in non-locomotive applications generally sit on a rigid base, while static beam deflections for North American diesel locomotives are about 0.6 in. It was not known at what point the engine's performance would be affected, but bending moments on the engine frame or drive train would have life and maintenance consequences. Thus to be effective, buff force reductions would have to be accompanied by engine weight reductions to limit the deflections associated with the more flexible frame.

Operation of a fuel tender involves a full set of capital, maintenance and transportation expenses. It also limits operational flexibility. If economies are derived from sharing a fuel tender with front-end, multiple-unit power, the benefits of distributed power are reduced. The use of dynamic braking must also be significantly reduced and more cautious train-handling techniques employed if a lightly loaded car is located at the front of the train. As improper train handling is one of the major causes of train derailments, the design of the tender has been assumed to be ballasted up to a 50 ton tare weight so that train handling would be treated the same as for existing express trains. One U.S. gallon/hp could be provided for a 12,000 hp consist on 70-ton trucks and a 50 percent range increase could be handled for a 15,000 hp consist on 100-ton trucks. The economic analysis assumed an operating cost of $0.18/mile for a tender.

7.2 Alternative Engines

High-Speed Lightweight Diesel Engine. The above values are estimates of the limits of conventional North American technology. There are many applications of existing technology in other countries where weight is of more concern. European diesel locomotives generally use a higher-speed engine (1,500 rpm) which has a higher power density. Weight-to-power ratios of 6.5 lb/hp are common and SEMT-Pielstick (Societe d'Etudes de Machines Thermiques) have developed an engine which achieves 5.5 lb/hp with the aid of a variable-geometry combustion chamber and a high-ratio turbocharger.[53] The higher speed also permits a reduction in the weight of the main generator if the traction motors can utilize a higher voltage. One means of accomplishing this is with ac traction motors. With ac traction, the motor supply is isolated from generator feed with intermediate energy storage devices. This permits a reduction in the current rating of the generator. However, it is of interest to note that the existing four-axle locomotive uses the same generator as the six-axle locomotive, and thus has a surplus current-carrying capacity of 50 percent. The economies of standardization from design and inventory savings are seen to outweigh the weight saving benefits of a smaller generator.

With higher engine speeds come higher fuel consumption rates, higher wear rates for liners and rings, and higher inertial forces. With a shorter time interval for combustion there is also less flexibility in the use of off-spec fuels.

Full-load brake-specific fuel consumption (BSFC) rates for 1,500 rpm turbochargecharged diesel engines are in the order of 0.365 lb/bhp/h.[54] Fuel consumption rates were scaled up from published data for a 1,500 rpm Rustin-Paxman engine.[55] For the EMD heavy-duty cycle, a 7 percent increase in fuel consumption results in comparison to a scaled-up, medium-speed diesel.

Maintenance data for medium higher-speed diesel engines in North American railway service are not available. For the economic assessment it was assumed that cylinder liners, fuel injectors and piston-related changeouts increase in proportion to engine speed. Thus, the frequency of engine overhauls and unscheduled maintenance on power assemblies was increased by a factor of 1,500/900 or 1.67. The cylinder-related costs of engine maintenance were isolated from the maintenance data base.

Assuming the same proportional reduction to fixed and variable components of the medium-speed diesel, the weight-to-power characteristic of the high-speed diesel engine is:

$$W\,(\text{lb}) = 4345 + 4.4\,P\,(\text{hp})$$

As noted, the generator for the four-axle locomotive has excess capacity. The weight of the power package of the high-speed, diesel-based locomotive must equal the weight of the medium-speed diesel being replaced (i.e. $W_{HD} + W_{HE} = W_{MD} + W_{ME}$). Using dc traction and substituting the appropriate values from Table 7.2, one has:

$$\begin{aligned}
W_{HD} &= 4345 + 4000 + (4.4 + 1 + 1.07 * (7.0 + 1.6)) * P_{HD} \\
W_{HE} &= 4 * (2000 + 5 * .225 * P_{MD}) \\
W_{MD} &= 7900 + 4000 + (8.0 + 1 + 7.0 + 1.6) * P_{MD} \\
W_{ME} &= 4 * (2000 + 5 * .225 * P_{MD})
\end{aligned}$$

Within the weight constraint of a 3,800 thp diesel locomotive, one can realize 4,600 thp by using a 4,800 hp high-speed diesel engine with dc traction motors. With ac traction there is sufficient capacity in the traction motors. Power-conditioning units are required at 1.5 lb/hp and savings of 0.75 lb/hp are realized with a higher-speed generator. Thus, the net change is:

$$\begin{aligned}
+ W_{HD} &= 4345 + 4000 + (4.4 + 1 + 1.07 * (7.0 + 1.6)) * P_{HD} \\
+ W_{HE} &= .75 * P_{HD} \\
- W_{MD} &= 7900 + 4000 + (8.0 + 1 + 7.0 + 1.6) * P_{D}
\end{aligned}$$

This results in a limit very close to the 4,600 thp realized with dc traction motors.

Diesel-Turbine Hybrid Engine. The turbine engine provides further improvement for lightweight locomotives. Its high power-to-weight ratio has made it a contender in high-speed passenger applications. Its principal drawback is its poor fuel economy, particularly when lightly loaded. The poor fuel economy of the simple turbine can be improved by using recuperators or regenerators for transfer of exhaust heat to the burner intake air, and the light load problem can be mitigated by cycling multiple engines to meet load demands. Other benefits of the turbine are its simplicity and flexibility in fuel acceptance. It does not require a separate cooling system, lubrication requirements are significantly reduced, and the complete unit can be readily exchanged for overhaul or maintenance.

The development of recuperators is important to railway applications. Early designs were primarily for fixed installations. They were bulky, heavy and required extended startup and shutdown periods to control the thermal and structural stresses. Recent developments of primary surface recuperators have led to significant size reductions. Solar turbines, for example, have a primary surface recuperator which is one-sixth the size and weight of shell-and-tube or plate-fin designs. It can also be taken from a cold start to full power in eight minutes.[56]

An extension of the idea of cycling multiple turbines to meet the operational duty cycle is to have a hybrid power plant with a diesel engine meeting the load fluctuations and a turbine brought on-line at peak load conditions. To achieve this, the turbine is coupled into a double-ended generator. This requires a turbine with separation of drive and compressor shafts or the use of an inexpensive and lightweight overrunning clutch. It would not be possible to have a separate turbine-mechanical drive on one of the locomotive trucks as traction demands at low speed require all axles to be powered. The double-shafted generator method of operation has the added benefit of extracting the maximum utilization of the generator's capacity. A traction generator must be designed to provide a high current at low voltages for low-speed operation and a high voltage at low current for high speeds. Consequently, its power capacity is under-utilized by the constant power parabola of the engine. The engine and generator characteristics of a typical diesel-alternator set are illustrated in Figure 7.1. The excess generator capacity which is available in the medium to upper speed range allows an incremental power of up to 25 percent without increasing generator capacity.

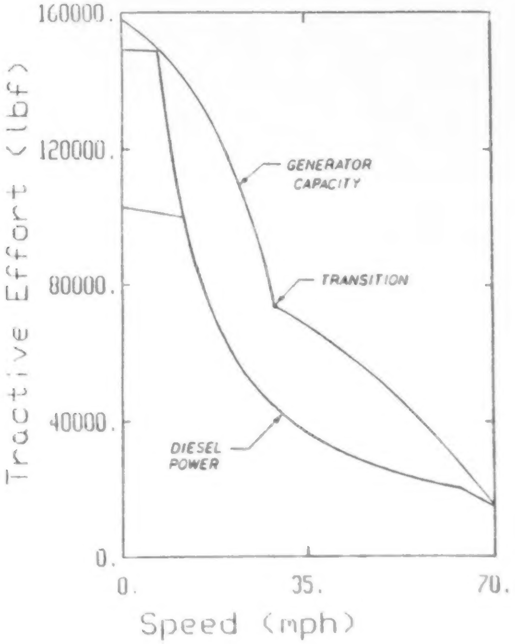

Figure 7.1 Engine and generator characteristics of diesel-alternator set

The weight-to-power ratio of a simple turbine is an order of magnitude smaller than that of a diesel engine. With a recuperator, gear reduction and intake and exhaust plenums considered, the weight-to-power characteristic is estimated to be:

$$W\ (\mathrm{lb}) = 700 + 2.19\ P\ (\mathrm{hp})$$

The main weight concern of the turbine engine is its fuel requirements. The recuperated turbine at its most efficient load has about a 12 percent increase in consumption over the diesel engine. From Table 7.2, the weight of the power package of the hybrid locomotive is:

$$RW = W_T + W_D + W_E$$

where: $W_T = 700 + (2.19 + 1.12 * (7.0 + 1.6)) * P_T$

$W_D = 7900 + 4000 + (8.0 + 1 + 7.0 + 1.6) * P_D$

$W_E = 4 * (2000 + 5 * .225 * (P_T + P_D))$

The weight constraint of a 3,800 thp diesel locomotive can be met with a 4,600 thp hybrid consisting of a 2,000 hp (eight cylinder) diesel and a 2,700 hp turbine.

The turbine output power is much more sensitive to temperature and elevation than is the diesel engine. Shaft power must be derated by about 3.6 percent/1,000 ft elevation above sea level and 0.511 percent per degree Fahrenheit above design temperature (usually 59° F).[57] These are power deratings which do not affect the energy efficiency of the engine. In addition, there are pressure losses in the intake filter system and losses in the gear reduction unit which do affect efficiency.

As the weight of the turbine is a relatively insignificant part of the total package weight, it can be increased in size to overcome the derating factors. There will, of course, be an economic penalty in purchase price. The engine selected for comparison provides the full 2,700 hp at 2,000 ft altitude and 80° F, with a 100 hp loss in intake air pressure loss and a 0.97 gear efficiency. On this basis, a turbine rated at 3,375 hp is required. Operation above these limits will result in performance loss. The duty cycle of each engine and the combined fuel consumption based on scaling of manufacturer's data with the above losses considered is shown in Table 7.4.

TABLE 7.4

DIESEL/TURBINE HYBRID FUEL CONSUMPTION

		Percent Rated Output			Fuel Consumption		
					Specific Rate (lb/hp-hr)		
Notch	Auxiliary Power (hp)	Loco	Diesel	Turbine	Diesel	Turbine*	Total Rate (lb/hp-hr)
Dynamic							6.50
Non-Op. Idle	0	0	0	0			4.40
Idle	0	0	0	0			6.23
N1	18	3	7	0	0.385		31.23
N2	35	9	21	0	0.365		85.23
N3	60	19	46	0	0.338		164.18
N4	90	32	77	0	0.328		266.45
N5	60	48	60	40	0.338	0.500	488.66
N6	90	64	90	46	0.328	0.470	608.68
N7	100	83	100	71	0.332	0.420	763.98
N8	100	100	100	100	0.332	0.369	863.48

* With gear losses of 3 percent and pressure losses of 100 hp included.

Another alternative is to eliminate gear reduction with a high-speed generator. Donnelly has suggested the use of such a device in a similar application.[57] The high-speed synchronous generator with a permanent magnet rotor is at the conceptual design stage. Indications are that the generator would weigh no more than the gear reduction unit it would replace. However, one would require a system to cool the compact stator windings at the current levels required, and a thyristor-based power-conditioning unit to interface with the diesel's main generator output. Energy losses, maintenance costs and first cost were not considered to be competitive with the 3 percent gear loss that it would save.

Maintenance Costs. Maintenance cost for railway applications of turbine engines is an area of relative uncertainty. Turbine deterioration is quite sensitive to the number of start cycles it experiences and the variation of power level while operating. Operators generally develop maintenance schedules from experience on the basis of either time between overhauls (TBO) or maximum start cycles. This may vary from over 20,000 hours for large constant power-fixed installations to 2,000 hours for helicopter service with 20 minute flight times. The aviation industry has traditionally had a regulatory limit on TBO of 3,500 hours. Recent shifts to on-condition maintenance has led to TBO of 6,000 hours for Pratt & Whitney PT6 engines with some fleet-leaders exceeding 20,000 hours.

Rail application of turbines adds further problems in the area of blade erosion from particulates in the intake air. This was a major concern for CN in the operation of the Turbotrain in Canada in the 1970s. The ANF Turbotrain encountered the same problems with desert sand in Iran. However, design changes to the filtering system for intake air solved this problem.[58]

Maintenance costs used in the evaluation of turbine engines were based on reported experience in similar service. This includes Amtrak's experience with the ANF Industries' Turbotrain and the Canadian Armed Forces' experience with Solar and Pratt & Whitney turbines used as booster engines in destroyers. In both cases, the maintenance requirements are much higher than those experienced in fixed installations. Amtrak undertakes an engine overhaul at the manufacturer's recommended interval of 4,000 hours. This is an improvement over early train applications of turbine engines which required overhauls at 1,500 hour intervals. Amtrak's full cost of overhaul varies by type of engine and is quite sensitive to parts costs and the exchange rate on the French Franc. The range was from US$100/kW to US$200/kW for each overhaul.

Turbine boosters on the Canadian destroyer fleet are run to failure. Engines are exchanged and the overhaul work is contracted out. Their 800 kW engine averages 3,400 hours between failures, with an average overhaul cost of $60,000 ($75/kW), while the 2,500 kW engine averages 2,500 hours between overhauls costing $115,000 ($46/kW).

The maintenance costs associated with the recuperator must also be considered. The primary surface recuperator which forms the basis of the hybrid locomotive weight calculations has a design life of 100,000 hours and 5,000 starts. Both have been exceeded in limited applications to date.[59] The governing factor in the hybrid application would be the number of starts. With usage at 50 percent power and above, the turbine would be on line two-thirds of the time, but may see an average duration of less than two hours. Thus, its life in locomotive service is less than 10 percent of its potential life in fixed plant usage.

The mean time between overhauls (MTBO) figures can be translated into years of service via the service cycles of Table 2.1 (line-haul utilization of 48.6 percent) and the express duty cycle of Table 2.5 (66 percent of the time above N-4). Therefore the hours/year on line is $0.486 * 0.66 * 8760 = 2810$.

The 2,500 kW destroyer application is closest to the size and duty cycle of the hybrid locomotive being evaluated. Its overhaul characteristics have been adopted as the basis for evaluation. In order to simplify calculations, the 2,810 hours annual service was taken as the MTBO and the cost/ overhaul was increased by 12 percent to compensate for the difference. The assumption for the recuperators was that they are replaced on every third engine overhaul (i.e. a 8,430 hour life corresponding to an average run time of 1.68 hours). These values contain a high level of uncertainty, so maintenance cost is an important parameter for sensitivity analysis.

One of the advantages of the turbine engine is that it can be changed out for overhaul rather than worked on "in place." Thus, locomotive availability is not significantly affected. Also, in the hybrid application the diesel engine has only eight cylinders. Maintenance costs for the diesel engine will be significantly lower than for the base-case locomotive.

Because the life of the recuperator is so low in the railway duty cycle, a second non-recuperator hybrid was evaluated. Without the weight of the recuperator, the 4,600 thp could be supplied with a 12-cylinder diesel and smaller turbine.

Purchase Price. The capital costs of turbine and recuperator were scaled from a range of budgetary quotations. The price for a two-shaft simple turbine complete with reduction gear is estimated at:

$$\$250,000 + \$188.45/\text{kW},$$

while the price for a recuperator is estimated at:

$$\$100,000 + \$175.74/\text{kW}.$$

The 2,000 kW output requirement can be met with a 2,518 kW turbine and a 2,000 kW recuperator.

7.3 Economic Comparison

The capital purchase value under a rebuild decision was derived for each alternative on the basis of the unit cost performance developed above. The results are presented in Table 7.5. In terms of $/tractive horsepower, each of the diesel options is in the same range.

TABLE 7.5

CAPITAL PURCHASE VALUE OF
ALTERNATIVE EXPRESS LOCOMOTIVES

Alternative Description	No. of Axles	Power (thp)	Value (M$/unit)	Value ($/thp)
Weight Reduction	4	3,850	2.28	592
High-Speed Diesel	4	4,600	2.80	608
Simple-Turbine/Diesel Hybrid	4	4,600	1.90	413
Recup-Turbine/Diesel Hybrid	4	4,600	1.18	256
Tender/16 Cylinder Diesel	4	5,750	3.81	660
Tender/24 Cylinder Diesel	4	5,750	3.74	650
Co-Co/24 Cylinder Diesel	6	5,750	3.45	600

The high-speed diesel engine realizes an estimated value in $/thp very close to the medium-speed diesel alternatives with separate fuel tender. Given the extensive experience the railways have with the medium-speed diesel, there would appear to be insufficient advantage to warrant a switch in technology.

Considering the medium-speed diesel alternatives, the value per locomotive is much higher at higher power levels. If economies of scale can be achieved in production such that the cost in $/thp decreases with increasing power, the 5,750 thp alternative may be a product of benefit to both builder and railway. The estimated difference in value for the 16- cylinder engine versus the 24-cylinder engine is not a significant one. This is based on the premise that the materials cost per cylinder is in direct proportion to its power rating and the MTBF remains unchanged on a per-cylinder basis. It is also evident that the six-axle version of the 5,750 hp locomotive is of the same value as the four-axle unit with separate fuel tender.

The diesel-turbine hybrids are significantly lower in value. This is due to the anticipated high maintenance cost and lower efficiency. Somewhat unexpectedly, the recuperated turbine hybrid is of less value than the simple turbine hybrid. The fuel consumption savings are not offset by the LCC of the recuperator in the railway duty cycle.

The sensitivity to the hybrid cost estimates was investigated. If the recuperator's cycle life were extended from 5,000 to 35,000 cycles, the locomotive value would double to $2.2 million, still only slightly higher than the simple turbine. If, in addition to this, the turbine maintenance costs were reduced to one-third of the estimate (from 2,500 hr TBO to a 7,500 hr TBO), its value would increase to $3.58 mllion. The same maintenance reduction to the non-recuperated turbine results in a locomotive value of $2.34 million. It is clear that the turbine engine will have to be demonstrated to be capable of achieving much lower maintenance costs than have been exhibited in similar duty cycles before it will be a candidate for railway express freight applications.

8. ALTERNATIVE FUELS

The cost development of Section 2 revealed the importance of fuel costs in the life cycle costs of a locomotive. Subsequent sections incorporated the value of reducing this cost factor as one component of the alternative technologies considered. This section deals with the value of alternative fuels. It is primarily concerned with diesel engine fuels, consistent with the current technology time frame of the study, but also considers the longer term potential of hydrogen as a railway fuel. The assessment is focussed on two fuels which have received recent attention from the North American rail industry -- natural gas and coal.

The economic evaluation of alternative fuels contains a much higher level of uncertainty than that associated with technological factors alone. Energy prices are subject to market factors much stronger than those of the railways alone. Historic price relationships can be significantly disturbed via substitution of fuels in other markets (transportation and non-transportation, domestic and international). In addition, existing fuel prices cannot always be compared, as they exist in different markets. Government taxes constitute differing proportions of market price for different fuels and for the same fuels in different markets. The conservative approach to fuel costing has been taken, with the assumption that those tax revenues which are currently derived from railway consumption of diesel fuel will be derived from consumption of alternative fuels. The railways consume a significant proportion of domestic diesel fuel and conversion would lead to a very noticeable reduction in tax revenues from refining and sales. The impact on crude oil consumption, on the other hand, would be negligible and would in turn have a negligible impact on well-head tax revenues. Thus all taxes on diesel fuel are assumed to be transferred to any alternative fuel, while crude oil taxes are not. This assumption may be overly conservative in the consideration of small-scale localized conversions, but is considered to be a realistic expectation of fact in regard to major substitutions of fuels.

8.1 Natural Gas

Natural gas is a relatively abundant resource in Canada. A large distribution infrastructure presently exists. Until recently it had a regulated price, set to a fixed ratio of the domestic oil price. A price ratio of 0.65, plus capital grant incentives, was used between 1981 and 1986 to encourage conversion of residential heating from oil to gas and, to a lesser extent, conversion of vehicles from gasoline to compressed natural gas (CNG). Over 5,000 spark-ignition vehicles in Canada operate purely on CNG. At higher power levels, natural gas has long been the fuel of choice for fixed-turbine applications. More recently, conversions have been made by utilities and marine operators of diesel-electric generating facilities from #2 diesel fuel to dual-fuel diesel/natural gas operations. Within the North American railway industry, the Burlington Northern Railroad is experimenting with a prototype dual-fuel locomotive in linehaul service.[60]

Since November 1986, all Canadian natural gas prices have been fully deregulated by the federal government and are freely negotiated between buyers and sellers. To date, many direct sales have been completed at significant discounts. At the same time, producers have been given greater access to export markets. This factor will have a growing importance on Canadian domestic natural gas prices.

Dual-fuel conversions, which require replacement of pistons, cylinder liners and cylinder heads, have been considered economical, given CNG prices and the prime-mover duty cycle involved. There are a number of deterrents to the application of CNG to railway motive power, the principal one being the need for portability of large quantities of fuel. While utility conversions can draw natural gas directly from the pipeline, railway conversions require compression or liquefaction to reduce to a manageable volume the gas required to propel a locomotive the same distance as its existing diesel fuel load. This introduces extra processing costs and facilities as well as safety concerns.

Another deterrent is the railway duty cycle. Using natural gas in dual-fuel, two-stroke diesel engines has efficiency and stability drawbacks at lower engine loads. Railway locomotives cycle frequently between low and high power outputs, and spend considerable time at idle.

The price premium paid for railway diesel fuel over the cost of crude oil includes the costs and markups of refining and distribution and the imposed federal and provincial taxes. The corporate factors are estimated at 27 percent, diesel taxes at 18 percent, and provincial railway fuel taxes at 2.6 cents per litre. The $0.286 per litre base-case cost of diesel fuel includes $0.066 per litre in taxes which are assumed to be transferred to natural gas.

The base case price of natural gas (NG) is estimated to be discounted 20 percent below the previous regulated price of 0.65 time crude oil price. This represents a raw NG price at 31.5 percent of final diesel fuel prices. Adding corporate costs and markups for distribution and compression (estimated at 10 percent) and transferring the taxes, compressed natural gas (CNG) is available at 58 percent of the price of diesel fuel.

If the NG is liquefied (LNG) to reduce fuel tender requirements, the cost of liquefaction must be allocated in place of the cost of compression. The budgetary cost estimates shown in Table 8.1 were provided by Intercity Consumer's Gas Co. for a liquefaction plant which they had recently considered building. The LNG cost implications of such a facility operating with an 8 percent real ROI are an incremental $0.0514 per diesel equivalent litre of NG. With 5 percent distribution cost and taxes transferred, the cost of LNG to the railways would be 74 percent of diesel fuel on an energy-equivalent basis.

TABLE 8.1

LIQUEFACTION FACILITIES COST ESTIMATES

Capacity (litres of D2 equivalent per annum)	161,000,000
Capital Cost (1986$)	74,000,000
Operating Cost ($/annum)	4,765,000

Another means of using NG as a railway fuel is through methanol production. Appendix C discusses some of the encouraging results obtained in methanol combustion in two-stroke diesel engines. Methanol production is more expensive than liquefaction of NG but achieves a higher displacement of diesel fuel and can use a lower-cost tender. The cost of methanol in relation to raw natural gas is derived from 5 percent distribution costs, 20 percent energy costs, and $0.06/litre plant costs of methanol synthesis. As methanol production generates tax revenue only, the $0.026/diesel-equivalent-litre provincial railway fuel tax is transferred. As methanol has a heating value of only half that of diesel fuel, two litres of methanol are required to displace one litre of diesel fuel. The above cost relationships produces a diesel equivalent price for methanol of 91 percent of diesel fuel.

NG has about a 10 percent higher energy content than diesel fuel, so weight is not a concern. However, The volumetric storage requirements of CNG and LNG are approximately 6.5 and 1.7 times that of diesel fuel. Thus, a separate fuel tender is necessary for CNG in the order of one per locomotive. LNG could be located on board the locomotive if the additional volume could be located. In a "B" unit, an 11 foot long, 8 foot diameter (inside dimensions) tank could be physically located in the cab space, providing about 3,850 U.S. gallons of diesel-equivalent fuel. Multiple tanks may be necessary on an "A" unit, thereby losing the economies of scale in a single tank. A separate LNG tender car could provide four to five times this storage volume. A tender would be required for methanol because twice as much fuel is required on an energy-equivalent basis while the density is about equivalent. One tender per train in combination with on-board fuel tanks would provide extended range.

Tender capital costs have been estimated as follows:

CNG tender	$300,000
LNG tender	250,000
LNG: one on-board tank	120,000
LNG: two on-board tanks	155,000
Methanol tender	60,000

As discussed in Appendix C, NG must be used in a dual-fuel mode in medium-speed diesel engines. On the basis of the Burlington Northern conversion, it is assumed that a 5 percent pilot injection of diesel fuel is required in conjunction with NG and that the use of NG is limited to higher power levels (notches 6, 7 and 8). In order to improve the utilization of NG, it is assumed that a load transfer policy would be followed where one or more units of the consist is operated in idle in order to allow the other units to operate in a higher notch where NG can be consumed. This was quantified in deriving fuel consumption estimates by assuming that 75 percent of the time spent in notches 4 and 5 was met by operating half of the units in notches 6 and 8, respectively, and the other half in idle. In the Burlington Northern conversion, the engine was derated in order to get a better match in power increments between diesel notch 5 operation and dual-fuel notch 6 operation. However, there appears to be no combustion limitation to force a derating of the engine in notch 8. A revised control system would be required to achieve a desired power level in each throttle setting. Engine power was not derated in our assessment of alternatives.

The diesel/turbine hybrid has an advantage in achieving high levels of fuel substitution. The turbine can operate on 100 percent NG and as it is on-line at higher notches only, it raises the diesel engine output power for lower locomotive notch settings, as illustrated previously in Table 7.4. The hybrid locomotive realizes a displacement of 95 percent of diesel consumption, where the diesel engine following the above shut down policy achieves an 88 percent displacement ratio. However, the maintenance costs of the hybrid have already removed it from contention. Of the other alternatives considered, those already using a fuel tender will realize a larger incremental benefit from NG conversion.

The capital purchase values for each of the NG-based conversions were derived on the basis of the same tender operating costs as the diesel fuel tender in Section 7. The results are summarized in Table 8.2. The values in the first column are based on a retrofit to existing SD40 locomotives. The purchase values are shown for both the locomotive conversion and tender purchase, as well as with the estimated tender cost factored out. It is seen in comparing LNG and CNG that the incremental costs of liquefaction are more than offset by the benefits of a reduced number of tenders. CNG permits only a modest allowance for conversion costs and R&D investment. The incremental processing costs of methanol are of sufficient magnitude to eliminate any fuel cost savings beyond those necessary to cover the incremental operating costs of the tender.

TABLE 8.2

CAPITAL PURCHASE VALUES OF NATURAL GAS CONVERSIONS*
(thousands $)

		Loco. Type	
Case	Base-Case Retrofit	4-Axle 5,750 hp Loco.	6-Axle 5,750 hp Loco.
CNG, with one tender/locomotive	$398 ($ 98)	n.a. n.a.	n.a. n.a.
LNG, with one tender/train	$428 ($345)	$4,780 ($4,697)	$4,220 ($4,137)
LNG, with one on-board fuel tank	$550 ($430)	n.a. n.a	$4,460 ($4,340)
LNG, with two on-board fuel tanks	$550 ($395)	n.a n.a	$4,460 ($4,305)
Methanol, with one tender/train	$ 53 (nil)	$4,240 ($4,180)	$4,680 ($3,620)

* Purchase values must cover tender capital costs where applicable, bracketed numbers are for locomotives only.

The R&D allowance associated with the base-case locomotive retrofit to LNG is illustrated in Table 8.3. In Table 8.4, the incremental allowance to develop an on-board LNG tank is presented.

TABLE 8.3

LNG CONVERSION OF SD40 LOCOMOTIVE

Capital Purchase Value (K$/locomotive after taxes) 345.00
R&D Allowance
(M$ after taxes with 25 percent ROI)

NPV to Railway	Cost of Installed Product (K$/loco.)				
(M$)	266	233	200	167	133
53.3	n.a	n.a	0	22	44
40.0	n.a	0	22	44	67
26.6	0	22	44	67	89
13.3	22	44	67	89	111

No. of Locomotives: 400

TABLE 8.4

INCREMENTAL VALUE OF ON-BOARD LNG TANK OVER LNG TENDER

Capital Purchase Value (K$/locomotive after taxes) 205.00
R&D Allowance
(M$ after taxes with 25 percent ROI)

NPV to Railway		Cost of Installed Product(K$/locomotive)				
(M$)	174	152	141	130	119	109
34.7	NA	NA	NA	0	7	14
26.0	NA	0	7	14	22	29
17.4	0	14	22	29	36	43
8.7	14	29	36	43	51	58

No. of Locomotives: 400

8.3 Hydrogen Fuel Cell

Fuel cell development has been largely funded by the U.S. space program. It may be considered a long-term alternative for railway applications.

Fuel cells resemble batteries, but are energy converters rather than energy storage devices. Oxidizers and fuels are supplied continuously to produce electricity at the terminals. Oxygen is used as the oxidant at the cathode, and hydrogen, which may be derived from electrolysis (basically the reverse operation), or fuels such as coal, natural gas and methanol. Conceptual designs of fuel cells have been developed for the locomotive application. A Transportation Research Board (TRB) committee[61] considered both phosphoric acid and solid polymer electrolyte fuel cells, while The Institute for Hydrogen Systems (IHS) in Canada assessed alkaline fuel cells.[62] Each was considered to be technologically feasible. Weight savings were realized in displacing the diesel engine and alternator but additional volume had to be found.

Reduced energy consumption is the principal source of savings to be derived with fuel cells. Expected efficiencies in the range of 55 percent would realize a fuel consumption of 70 percent of those achieved with a diesel engine at 39 percent efficiency. However, the cost of producing the necessary hydrogen fuel more than offsets this efficiency advantage.

The IHS report estimated the cost of producing liquid hydrogen from electrolysis and methane reformation at $37.59/gigajoule and $20.45/gigajoule, respectively. Liquid hydrogen from electrolysis was estimated on the basis of electricity at $0.031/kWh and results in a price of $1.41/diesel-equivalent litre. This is about five times the cost of diesel fuel. The 0.7 consumption advantage of the fuel cell would reduce the effective cost ratio to 3.5 times diesel. About half of the cost of hydrogen production is estimated to be directly tied to electricity costs. Even if electricity were available at no cost, the capitalization and other operating costs would still exceed the diesel equivalent price. If off-peak nuclear generating capacity were used to produce hydrogen, as has been suggested by many proponents of hydrogen fuel, it would result in lower utilization of the electrolysis plant investment and may more than offset the benefits of lower electricity prices.

The IHS-estimated cost of hydrogen from methane synthesis (corresponding to $0.76/diesel-equivalent litre) was based on a natural gas price at $0.15/diesel equivalent litre or 53 percent of our base-case diesel price. On the basis of the NG price assumptions made in this present study, their estimate would be reduced to $0.625. With the fuel cell efficiency advantage, the final energy cost of the hydrogen fuel cell is 1.53 times the cost of diesel fuel.

The TRB report considered locomotives with on-board methanol reformation. In this situation, the cost of hydrogen liquefaction is eliminated, but would be offset to a certain degree by lost economies of scale in reformation. The net present value of a fuel cell with on-board methanol reformation was assessed on the basis of 70 percent diesel-equivalent consumption of methanol and one tender per train, to be $520,000. This fuel saving when combined with the capital and operating costs of the diesel engine and alternator (which would be eliminated), corresponds to a life cycle allowance for a fuel cell and reformer of 1.9 times that of the diesel engine and alternator. There are very little data published on either the capital or operating costs of fuel cells. Price ranges of $300/kW to $1200/kW (1.4 to 5.9 times that of a diesel/alternator) and a life expectancy of three years for the platinum catalyst and ten years for the fuel cell have been noted. With the capital cost of reformer and the system maintenance costs included, the fuel cell alternative is not going to be competitive for even the low end of the capital cost estimates.

8.4 Coal

The development of coal as a railway fuel has received a considerable level of funding by U.S. industry and government in the 1980s. General Motors and General Electric each have developed conceptual designs for diesel engines burning coal-water slurrv and turbines fueled with either gasified coal or coal-water slurry. Millions of dollars have been invested in combustion research and in determining the economics of coal-fueled alternatives. It is not the intent of this study to duplicate the analysis performed by the various organizations.

The status of the technical research is reported in annual meetings sponsored by the U.S. Department of Energy. The 1986 meeting[63] revealed that much of the effort to date in regard to the diesel engine has been directed towards achieving satisfactory combustion through combustion chamber laboratory tests. This research will continue. The major problems of wear and erosion of diesel engine parts have not yet been addressed. Preliminary estimates of the maintenance impacts on the mature product are overhaul times of one-third to one-half those of the existing diesel-fueled engine experience and scheduled oil changes and injector replacements at two to three times the present frequency. The economic assessments of the proposed internal and external combustion coal alternatives have been summarized in a 1986 Department of Energy report[64] and GM and GE have presented assessments of their conceptual designs in separate papers.[65,66] This study has drawn on these reports for coal benefication and slurry costs.

The relationship between coal and diesel fuel prices is very similar to that of NG and diesel fuel. The cost of developing a coal-based railway fuel system are developed in Table 8.5. Raw coal which could be used in external combustion engines can be provided at about 36 percent of the base-case diesel fuel price. Once the diesel taxes are transferred, this figure becomes 60 percent. If further processing is undertaken to provide a coal-water slurry for internal combustion engines, the ratios become 67 percent before tax and 90 percent after transfer of taxes. In comparison to natural gas, lower fuel savings would be realized and incremental maintenance costs must be accounted for. A 50 percent increase in engine maintenance costs would fully eliminate the 10 percent fuel savings expected. In addition, the development costs and risks are much higher with coal and a completely new distribution system would be required. For these reasons, the coal alternatives were not pursued any further. They are not considered to be competitive with natural gas in Canada.

TABLE 8.5

COAL FUEL COST COMPONENTS

Component	Nominal Cost	Energy Cost ($/MBTU)	Cumulative Cost ($/MBTU)	(% of D2)
Mine Site Coal	$40/ton	1.48	1.48	19
Transportation	$22/ton	0.81	2.29	29
Distribution	$15/ton	0.57	2.86	36
Processing Plant		1.33	4.19	53
Processing Costs		1.13	5.32	67
Diesel Taxes		1.84	7.16	90

8.5 Electricity-Based Hybrids

Many forms of hybrid propulsion have been used in transport applications. Battery, flywheel and hydraulic accumulators have all been tested in transit applications where the frequent stop/start duty cycle offers an opportunity for recovery of kinetic energy of the system through regenerative braking. Battery storage has also been used to extend the range of electric rail cars beyond the limits of the distribution system. However, a brief review of advanced battery capabilities quickly led to the conclusion that the use of energy storage devices in Canadian rail freight operations will not be technologically attractive for the remainder of this century. Their power densities are below that of a diesel engine and their cost and cycle lives do not permit an economic means of storing electrical energy for diesel fuel displacement.

8.6 Dual-Mode Locomotives

The diesel/electric hybrid locomotive was proposed by the FRA and evaluated by AirResearch Corp. in 1981.[67] The advantages of the system were considered to be an improved utilization of surplus capacity in the dc traction motors, selective electrification of high fuel intensity track segments and a low front-end cost through incremental installation of the overhead contact system. In 1985, Bombardier assessed the potential for this particular form of hybrid and undertook layouts of conceptual designs.[68] A six-axle locomotive with a 12-cylinder 3,000 hp diesel engine with the necessary transformer and power-conditioning units for straight electric operation at the rating of the traction motors was considered to be feasible.

New York City's Metropolitan Transit Authority (MTA) has operated a "low-tech" version of this form of dual-mode locomotive since 1957. The FL9 units with a 12-cylinder, 2,000 hp diesel engine and resistance control of 600 volt third rail pickup were purchased to enable the same locomotives to continue through tunnels under the Hudson River where diesel engines cannot be operated.[69]

In 1986, MTA ordered ten dual-mode propulsion packages from BBC for a complete rebuild of the existing FL9 hybrids. The replacement units will use a 10-cylinder, 3,000 hp diesel engine, a 600 V to 1,400 V step-up converter for third rail pickup. The traction package uses ac traction motors with GTO-based pulse width modulated inverters. The axle arrangement is a Bo A1A, five axles to distribute the load weight with four powered. The unit is rated at 1,865 kW in electric mode. The price of the ten units, which may be considered prototypes, is US$43 million.

The conclusion which can be drawn from this order is that the price for a dual-mode locomotive is significantly higher than either a straight electric or diesel-electric locomotive.

The translation of this price into an estimate for mainline freight application requires a number of adjustments. The "learning curve" factor for an order of 100 units was estimated to be 0.8 and the net effect of removing the step-up converter and adding a transformer rectifier for an overhead feed operation was estimated to result in a credit.

The attractiveness of a permanent dual-mode operation was assessed. Incremental investment is required in a wayside distribution system for each electrified segment and in the locomotives to be operated in dual mode. The displacement of diesel fuel with lower cost electricity is the principal source of savings. In comparison to full electrification, the fuel savings per mile of fixed overhead contact system (OCS) is increased. However, the miles of OCS per substation decrease as segment lengths are shorter than the feasible feed lengths and locations are spread over the complete route. In the mountain region, upgrades are generally the most energy-intensive sections. Thus, the locations of peak fuel consumption for one direction correspond to the minimum fuel consumption for the other direction. The other complication for the study region is that the traffic splits at Golden. For the economies of the lower powered diesel engine to be realized, segments of the lighter density track segments must also be electrified.

The value of a dual-mode locomotive was evaluated for the mountain region. The operation was assumed to follow a strategy of diesel power for all segments of less than 75 percent of peak power and electric power for the remaining segments. In this way, a 3,000 hp diesel engine could be installed on a locomotive of 4,000 hp tractive capacity (consistent with Bombardier's conceptual design).

Considering the bulk trains under the assumptions of Table 8.6, the purchase value per locomotive before any fixed costs are considered is $3.3 million on a rebuild decision and $4.35 million on a buy-new decision.

Given the price estimates for a dual-mode locomotive there, is very little left for fixed installations and a railway portion of the NPV. As bulk trains are the dominant traffic class in this region, this concept was pursued no further.

TABLE 8.6

DUAL MODE LOCOMOTIVE PERFORMANCE

Adhesion	26 percent
Power (diesel hp)	3,000
Power (electric hp)	4,000
Engine Maintenance Savings*	57 percent
Electrical Maintenance Premium	56 percent
Electricity Consumption Charge	1.5 cents/kWhr
Electricity Demand Charge	$6/kVA/month
Average Load Factor	30 percent

* based on four less cylinders and 38 percent less run time

9. CONCLUSIONS AND RECOMMENDATIONS

9.1 Locomotive Service Distribution

Bulkhaul service receives the largest allocation of mainline motive power in Canadian railway operations (about 45 percent). Express service receives an additional 27 percent, and the remainder is in general freight service. This distribution is expected to continue in emphasis of bulk and express operations in the future. While there are numerous services where bulk unit trains are justified, there are very few where dedicated express unit trains can be economically operated.

9.2 Cost Components

Energy costs are the single largest cost component in the full life cycle cost of a diesel electric locomotive. For the base-case locomotive, it is 60 percent higher than the capital cost of the unit. Maintenance and overhaul expenses are about equal in present value to the capital purchase price of the unit. Within the area of annual maintenance and reliability costs, dc traction motors are the most significant cost components.

9.3 Component Retrofit Developments

Many of the alternative component developments evaluated in this report have been presented in terms of capital purchase value. This was deemed to be the most useful format to provide both railways and suppliers with the findings. In most cases the costs of alternatives are not available and are subject to varying market conditions and corporate ROI requirements. For this reason it is difficult to draw concrete conclusions on the economic attractiveness of specific component changes. It is recommended that the capital purchase values and R&D allowance tables generated in this report be circulated to the appropriate manufacturers and railway departments in order to stimulate discussion and firm up the recommendations which have been established here on the basis of rough estimates of capital and development costs.

The analysis of component improvements revealed that there were very few which would be economical on the basis of a retrofit to existing locomotives. Most component improvements must contribute to the enhanced locomotive performance in terms of tractive effort or power before they become economical.

The exception to this is the area of energy efficiency improvements. Ongoing research in the area of off-spec fuels should continue. In terms of hardware development, an opportunity may exist for Canada's only diesel engine manufacturer. The retrofit of the Bombardier four-stroke engine with a steam bottoming cycle looks attractive. Over 8 percent fuel savings can be realized with the conversion of two of its sixteen cylinders to a steam Rankine cycle. It is recommended that a detailed cost analysis be made of such a retrofit. The main area of development required to realize such a bottoming cycle is the engine load control system, which must incorporate the thermal time delays of the steam cycle with the loading demands of the main generator output.

There are also opportunities for Canadian companies to play a part in the development of electronic fuel injection systems.

9.4 Performance Enhancement

Analysis revealed that, while current-generation locomotives demonstrate a significant incremental value over their predecessors in a buy-new comparison, they compare very closely in a buy vs. rebuild decision. The situation is different in the U.S. where faster depreciation allowances make new locomotives more attractive in a buy vs. rebuild comparison. In Canada, improved performance is required to show a clear edge over the rebuild option.

In bulk service, tractive effort enhancement alone is sufficient to make new units attractive. This is a consequence of recent line capacity enhancements made by the railways and the very low cost of track-induced maintenance in comparison to other locomotive cost savings.

A coordinated program of research is recommended for tractive effort enhancement and is allocated the highest priority. The assessment of Section 6 identified opportunities in a number of areas.

The lowest cost, most readily implementable alternative is the substitution of alumina for sand as a friction-enhancing agent. Raising available friction would allow higher tonnage allocations for bulk trains on CN's complete system and on all but the mountain region of CP's mainline system. In CP's mountain region the controlling grades are of sufficient length that they cannot be climbed within the short-term thermal overload capacity of the traction motors. Laboratory and/or field evaluations of alumina are recommended to confirm the qualities reported by Japanese researchers and to assess the relative wear implications. These could be performed on the wear rig at the NRC Mechanical Division's Western Labs. In addition to these performance assessments, its flowability and ballast contamination implications need to be compared with those of sand.

Also related to friction enhancement, test results of the new wheel metallurgy developed by ASEA for railway wheels should be monitored. If the manufacturer's expectations of high friction performance are demonstrated, it is recommended that a wheel be compared with the results of the alumina tests under the same laboratory conditions.

The analysis in this report is based on computer simulation. It is recommended that field testing be undertaken to assess the adhesion performance in curves of dc and an ac traction motor locomotives and that a simple retrofit truck interconnection also be tested in order to validate the simulation based-findings of this study. CP Rail's prototype ac locomotive would be an ideal candidate for such an evaluation.

The interaction of mechanical and electrical design has been demonstrated to offer a range of alternatives to improved tractive effort performance. Only the supplier can assess which combination of mechanical and electrical technology can meet the performance attributes while providing a satisfactory return on development and manufacture expenses. In making this assessment, it is important to include the benefits of individual component improvements developed in Section 4 (e.g. steerable trucks and ac traction motors) with the benefits of enhanced tractive effort developed in Section 6.

While many of these alternatives would be met by existing offshore suppliers, new truck development is seen as an area of opportunity for Canadian suppliers. The focus of this development, however, depends on the choice of traction motor and must therefore be coordinated closely with a locomotive builder. Improved weight transfer is desirable in conjunction with dc traction motors, while reduced angle of attack in curving is beneficial to asynchronous ac traction. Individual axle control would largely eliminate the traction-enhancing benefits derived with these mechanical improvements.

In express service, higher powers are necessary to demonstrate a clear benefit in a rebuild comparison. A number of alternative lightweight engines was evaluated, and the conclusion was reached that the medium-speed diesel, although not the best in power/weight ratio, is the best choice from a LCC viewpoint to achieve high engine powers. The high-speed diesel engine realizes an estimated value in $/thp very close to the medium-speed diesel alternatives with separate fuel tender. However, given the extensive experience the railways have with the medium-speed diesel, there would not appear to be enough of an advantage to warrant a change in technology.

A diesel-/turbine-electric hybrid is not competitive with the straight diesel options. The turbine engine will have to be demonstrated to be capable of achieving much lower maintenance costs than have been exhibited in similar duty cycles before they will be a candidate for railway express freight applications. Recuperators will also have to achieve much higher thermal cycle lives to be a candidate in the rail application.

The development of special-purpose locomotives (i.e. bulk versus express) leads to smaller product markets from the builder's viewpoint and lower utilization from the railway's viewpoint. There is a definite value in a universal locomotive. This is now achieved with the present locomotive performance levels. However, enhanced performance in tractive effort cannot be achieved with a dc traction motor while meeting high-speed express train requirements. It is recommended that the penalties of specialization be further investigated. Two areas of development are required -- the gathering of data on locomotive cycle times in a range of service applications and yard sizes, and quantification of the policies used in dispatching locomotives to trains. The operations analysis tools which would be developed in this process will also be valuable in the assessment of means of increasing locomotive utilization levels above the present 50 percent.

Two alternatives maintain flexibility while achieving higher performance levels. The first is to use extra powered axles on adjacent coal cars in bulk service to realize the additional tractive effort with dc traction. Further investigation of this option is recommended. The series connection of dc traction motors results in a lower adhesion capability on a per-motor basis. The 66 percent increase in powered axles is only expected to provide a 40 percent increase in tractive effort. But this is still a significant increase. The technical concerns which must be addressed in the development of such a powered car are: a device which provides some mechanical advantage in coupling and uncoupling the electrical power cables between the locomotive and powered trucks on the adjacent car, a grade of dc commutator brush which will perform well when there is seldom electrical current being passed through it, location and fastening of control and air filtration equipment such that rotary dumping of the car and thaw shed heating do not damage components.

The second is to employ the inherent wide constant power range of ac traction motors which permit both high speed and high tractive effort operation. AC traction does not offer sufficient advantage in a retrofit without enhanced performance but may be attractive if it achieves high tractive efforts. The value of ac traction motors in maintenance cost savings is about $200,000, which is about half the market price. The value of a 13 percent increase in tractive effort for bulk service locomotives is over $700,000.

If a new traction motor development is necessary to achieve higher tractive efforts, it is recommended that the development expenditure be focussed on ac rather than dc technology. The benefits are close to covering costs and cost trends continue downward for power-conditioning units. There are a number of choices in ac traction systems and none are produced domestically. It is recommended that the maintenance experience with ac synchronous motors be obtained and the feasibility of permanent magnet fields be investigated. The main opportunity for Canadian firms is in the area of creep control systems for ac traction to achieve maximum adhesion performance in all conditions.

9.5 Alternative Fuels

The benefits of alternatives to diesel fuel are tied to differential price escalation. R&D expenditures in these areas are investments in risk reduction, providing alternatives in the event of rapid diesel price escalation. None are considered sufficiently attractive to warrant conversion at this time.

Natural gas was evaluated in compressed and liquefied form and as reformed methanol. Only the liquefied form is considered attractive as a conversion candidate. This is particularly true if the increased fuel volume requirement could be met on board the locomotive rather than in a separate LNG tender. CNG is the only near-term feasible candidate for a prototype. It may offer a sufficient return in a diesel retrofit application to warrant prototype development as a means of assessing the performance of the diesel engine and to obtain operating experience with a fuel tender. The objectives of a prototype demonstration would be:

(1) to achieve full engine rating under dual-fuel operation and achieve the desired power output at each notch setting regardless of whether the engine is dual-fuelled or straight diesel fuelled,

(2) to determine the lower limit of effective dual-fuel operation under different temperature and elevation conditions

(3) to quantify the operations and maintenance costs associated with a fuel tender.

It is also recommended that the feasibility of on-board LNG tanks be investigated. Layout concepts, safety concerns and regulatory restrictions need to be investigated.

Methanol does not provide sufficient fuel cost savings to offset the added costs of operating a tender. Coal-water slurry offers similar fuel economies to that of methanol, but there are still significant maintenance cost uncertainties. These factors, combined with the existing development efforts in the U.S., give it a low priority in Canadian railway alternatives.

Another advantage of a natural gas prototype locomotive is that it addresses many of the operating concerns associated with the use of liquefied hydrogen fuel tenders. Hydrogen could become a viable alternative in the longer term, although it would require significant increases in diesel fuel prices. Fuel cells appear to be technically feasible in the locomotive size range and are projected to realize continued increases in power density. While the capital and maintenance costs of a railway-sized fuel cell are not known, the economics of fuel cells is dependent on the cost of hydrogen. Hydrogen will always be more expensive than its energy source, be it electricity through electrolysis or natural gas through methane reformation. These costs are mitigated somewhat by the high fuel efficiency of the fuel cell, but are not competitive with diesel fuel at current prices. The use of off-peak nuclear generator capacity to produce low cost hydrogen is not considered to be a situation which will be available to all hydrogen consumers on a long-term basis and may be more than offset by the lower utilization levels for the electrolysis plant.

A dual-mode electric/diesel-electric locomotive would achieve much higher fuel savings per mile of electrified track than would full electrification but has other disadvantages. For bulk trains in the mountain region the purchase value equivalent of operating cost savings per locomotive, before any fixed costs are considered, was below the existing and projected price of a dual-mode locomotive. It is concluded that dual-mode operation will not be a viable alternative unless unusual local circumstances affect the cost savings.

A number of findings in this study are relevant to future electrification evaluations. The diesel-electric locomotive is considered to remain the basis of comparison for the remainder of this century. Either the SD40 rebuilt locomotive or the current-generation locomotives defined here may be used. The NPV of electrification assessments should be compared on a per-locomotive basis to the values attributed to the enhanced performance diesel-electric locomotives assessed in this study and estimates of development expense compared with the front-end fixed investments associated with electrification.

It is also important to recognize that the diesel engine has some flexibility in fuel usage. While significant improvements in energy efficiency are not seen, it is inappropriate to assume rapidly escalating diesel fuel prices as a basis of energy costs. Substitution of natural gas should be a standard basis of comparison in high fuel cost escalation scenarios.

From the electric locomotive viewpoint, the tractive effort enhancement alternatives considered in this report should be investigated. With its high power/

weight ratio, the value of tractive effort enhancement will be even higher for electric locomotives than for diesel electrics.

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APPENDIX A

MAINTENANCE COST ANALYSIS

Editor's Note: This appendix contains proprietary data and therefore is not included for general distribution.

APPENDIX B

INDUCED TRACK MAINTENANCE COSTS

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APPENDIX C

REVIEW OF FUTURE PERFORMANCE IMPROVEMENTS IN RAILWAY LOCOMOTIVE ENGINES

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Annex A: PERFORMANCE OF THE CONVENTIONAL ENGINE, WITH AND WITHOUT BOTTOMING CYCLES, AT DIFFERENT THROTTLE NOTCH POSITIONS

Annex B: PERFORMANCE OF THE LHR ENGINE, WITH AND WITHOUT BOTTOMING CYCLES, AT DIFFERENT THROTTLE NOTCH POSITIONS

C.1 FUNDAMENTAL DIESEL CYCLE IMPROVEMENTS

C.1.1 Introduction

The diesel engine has the highest thermal efficiency among the three types of internal combustion engines in common use. This is achieved mainly due to its ability to operate at a high compression ratio and thereby ensure that heat addition in the cycle occurs at a high temperature. The dictum for achieving a high conversion efficiency in any heat engine is to ingest thermal energy at the highest possible temperature and reject it at the lowest possible temperature. A great deal remains to be learned about the actual process of combustion inside the diesel engine. In spite of this, the high levels of performance and reliability obtained have been achieved by dint of development work over a number of years. A recent example of such development activity is the evolution of the Perkins "Quadram" combustion chamber for truck engines, which is reputed to achieve better air utilization by a proper matching of the air movement and fuel injection in the combustion chamber. However, such a combustion chamber may not be of great significance for locomotive engines with their high boost pressures, where the excess air factor for combustion is already quite high. At the present state of the art, the point of diminishing returns has long been crossed, and large improvements in diesel engine performance are hard to achieve by mechanical refinements alone.

The increase in oil prices during the 1970s and early 1980s, the uncertainties about petroleum supply and the general awareness about the depletion of readily available reserves have encouraged renewed attempts to achieve higher energy conversion efficiencies in powerplants. These factors have also promoted investigations of alternative fuels for use in these engines.

A comprehensive, in depth report summarizing these two aspects as they concern railroad locomotives was prepared at the end of 1981 by the Jet Propulsion Laboratory (JPL), California Institute of Technology, under the aegis of the U.S. Department of Energy [1]. An attempt will be made in the following to re-examine, critically, some of the conclusions arrived at in the JPL report.

C.1.2 An Overview of the Current R&D Effort in Diesel Technology

As the thermal efficiency of the diesel engine alone has more or less peaked, one has to consider other ways of achieving higher energy conversion efficiencies by suitable modifications of or additions to the basic diesel powerplant. A well-developed, modern, medium-speed diesel engine, of the type used in locomotives, converts about 38 percent of the energy input in the fuel to useful work and rejects most of the rest as exhaust enthalpy and as heat carried away by the coolant and the lubricant. A small amount of the energy input is also lost by the engine directly to the surroundings. The heat rejected in the exhaust amounts to as much as 35 percent of the total input. A part of the energy carried away in the engine exhaust can, in principle, be recovered as useful work by operating a heat engine with the exhaust gases providing the heat source. The higher the temperature of the exhaust gases, the higher the effectiveness of this "bottoming cycle." A part of the energy normally lost to the coolant can also be recovered by insulating the working space of the engine by the use of parts made of ceramics or other materials of very low thermal conductivity. A part of the energy retained by the working medium can be converted directly to useful engine output, while the remainder results in an increase in enthalpy of the exhaust gases. The bottoming cycle recovers more of this "retained" energy, thereby increasing the overall energy conversion efficiency of the entire system. This line of reasoning has led to a considerable amount of work being carried out by a number of organizations all over the world on the development of the so-called "adiabatic" engine. The result has been a great proliferation in the number of technical papers published on this subject. The Society of Automotive Engineers (SAE) has been bringing out annual collections of the more significant of these papers over the past few years [2,3,4] which are useful in obtaining an overview of the state of the art in this field.

It is impossible to design a truly adiabatic engine. Even if a perfect insulating material of zero thermal conductivity were found and used to cover the working space of the engine, it could only eliminate the convective heat loss from the cylinder charge. There is no way of eliminating the radiative heat loss. Unfortunately, the perfect insulating material has not yet been found, and the best that can be achieved at the present time appears to be a reduction of between 40 and 60 percent of the convective heat loss from the cylinder charge. As the results of a computer simulation of the diesel cycle to be presented later show, even a reduction of 50 percent of the convective heat loss from the charge entails an average temperature of 1,000°K for the working surfaces of the engine. This presents formidable material and tribological problems to be overcome in engine design and operation. Under these circumstances, it is more appropriate to call these engines Low Heat Rejection engines (LHR engines).

One of the pioneers in the development of LHR engines is the Cummins Engine Company of Columbus, Indiana. They have been developing an LHR engine for the U.S. Army for application in tanks [5] for a number of years. The argument is advanced that the elimination of a liquid coolant circuit and the attendant heat exchanger renders the tank less vulnerable to damage by enemy attack. These considerations are not particularly relevant to locomotive engines. Engineers at the Cummins Engine Company have carried out a number of theoretical analyses to predict the ultimate capabilities of the LHR engine. They have considered all possible ways of converting the energy retained by the working medium to useful work. These include turbocompounding and bottoming cycles. Their prediction is that it should be possible to attain a Brake Thermal Efficiency (BTE) of as high as 63 percent.

The JPL report [1] appears to accept the Cummins predictions for the performance of an LHR engine without reservations. However, we feel that these predictions have to be viewed with caution, if not skepticism. Our cycle simulation computations of a turbocharged locomotive engine, with 50 percent reduction in convective heat losses, show that the average charge temperature during the combustion period is about 1,650°K. It is reasonable to assume that the minimum temperature to which those gases can be cooled before exhausting is about 400°K without incurring the penalty of excessive heat exchanger surface areas, particularly in transport applications. At these temperature levels, the maximum theoretical efficiency of energy conversion (Carnot Efficiency), as dictated by the Second Law of Thermodynamics, is itself about 75 percent. It is hard to accept that up to 85 percent of this theoretical maximum conversion efficiency can be achieved by any practical device.

C.1.3 Computer Simulation of a Locomotive Engine

Some results of a computer simulation of a locomotive engine built in Canada (the Bombardier engine) are presented in this section. It is not possible to provide all the details of the diesel cycle model used. It is a single zone model which takes into account the variations with temperature of the thermodynamic and transport properties of the working medium. It incorporates a new heat transfer model developed by two of the authors of this report to predict the heat transfer between the working medium and the confining walls [6]. The model includes a heat release pattern that can be adjusted to simulate the heat release pattern in the engine under consideration. The agreement between the model predictions and the performance characteristics of the Bombardier engine over the closed part of the cycle was good, as shown in Figure C.1 and Table C.1 [7]. This encouraged us to use the model to predict the performance of the Bombardier locomotive engine, together with modifications and additions intended for improvements in thermal efficiency, over the whole locomotive duty cycle.

Some of the data required as input to the computer model were supplied by Bombardier several years ago. These data may be somewhat dated now as a result of their ongoing development work on the engine. However, the general nature of the results, and the conclusions derived therefrom, are not affected by this. It is a simple matter to repeat the computations if the latest data become available.

TABLE C.1

COMPARISON OF MODEL PREDICTIONS WITH MEASURED DATA — CLOSED PART OF THE CYCLE

	Model Prediction	Measured Value
Indicated Mean Effective Pressure (psi)	255.5	254.6
Indicated Thermal Efficiency	0.36	0.353
Maximum Cylinder Pressure (MPa) at Crank Angle	12.3 at 372°	12.34 at 371°
Maximum Cylinder Temperature (°K) at Crank Angle	1729° at 390°	1750° at 387°

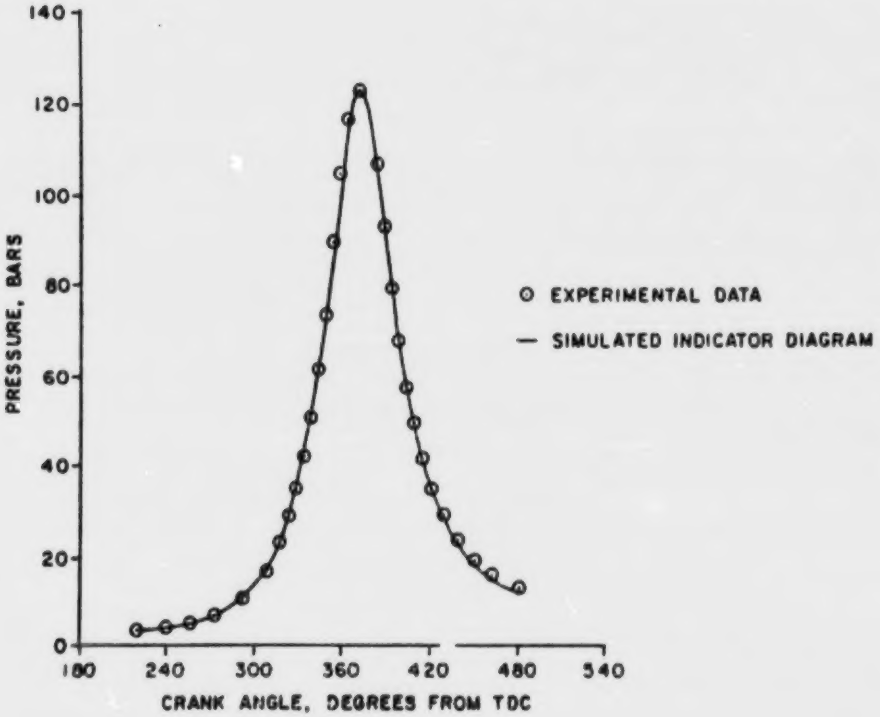

Figure 1 Predicted and measured indicator diagrams for the Bombardier engine

The following hypothetical versions of the Bombardier locomotive engine were investigated:

1. (a) The basic turbocharged, water-cooled engine (TC)

   (b) The basic engine with a Rankine bottoming cycle (TC+R)

   (c) The basic engine with a Stirling bottoming cycle (TC+S)

2. (a) A low heat rejection version of the basic engine, with the convective losses reduced to 50 percent of those in the conventional engine (LHR)

   (b) LHR engine with a Rankine bottoming cycle (LHR+R)

   (c) LHR engine with a Stirling bottoming cycle (LHR+S)

The performance of each of these versions was estimated with the help of the computer model over a typical heavy-duty cycle of the locomotive. The details of this duty cycle are listed in Table C.2.

TABLE C.2

TYPICAL LOCOMOTIVE HEAVY-DUTY CYCLE

Throttle Position	Engine Speed (% maximum)	Output (% of full load)	Time of Operation (% total)
Notch 1	33	5	3
Notch 2	43	12	3
Notch 3	52	23	3
Notch 4	62	35	3
Notch 5	71	51	3
Notch 6	81	66	3
Notch 7	90	86	3
Notch 8	100	100	30
Idle	33	0	41
Dynamic braking	-	-	8

Before the cycle simulation could be carried out, it was necessary to make certain assumptions about the engine and the different components of the entire system. These are described below:

(a) The friction losses in the engine were estimated according to a method suggested by Taylor [8]. Following this method, the Friction Mean Effective Pressure (fmep) is calculated according to the relation:

$$\text{fmep} = (\text{fmep})_0 + x(p_e - p_i) + y(\text{imep} - 100.0)$$

where: $(\text{fmep})_0$ = friction mean effective pressure (psi) of the engine under normally aspirated conditions, usually determined by motoring tests

imep = indicated mean effective pressure, psi

p_e = exhaust manifold pressure, psi

p_i = inlet manifold pressure, psi

x,y = correction factors to account for the difference between motoring and running conditions

The factor $(\text{fmep})_0$ depends on the type of engine and the mean piston speed. Factor x depends on the valve flow coefficients and factor y on the mean piston speed. Taylor has presented charts developed on the basis of experience with a large number of practical engines, from which $(\text{fmep})_0$, x and y can be estimated.

(b) The Bombardier engine uses a turbocharger with a fixed area nozzle. The boost pressure therefore varies with engine speed and load. In the absence of exact data, it was assumed that the boost pressure and the pressure drop across the turbine both vary linearly with engine speed. Such a trend obtains in turbocharged engines generally [10]. In addition, an average isentropic efficiency of 75 percent was assumed for the turbine to calculate the temperature drop across the turbocharger and hence the exhaust stack temperature.

In this analysis, no attempt was made to consider the effects of turbocompounding because it was felt that it would be a formidable task to transfer power from the power turbine to the engine crankshaft, given their different speed ranges over the duty cycle.

(c) The effectiveness of the bottoming cycle was assumed to be 20 percent with a Rankine cycle engine and 30 percent with a Stirling cycle engine. The effectiveness of the bottoming cycle is defined as:

$$\text{Effectiveness} = E_b/E_c$$

where: E_b = energy conversion efficiency of the bottoming cycle engine

E_c = Carnot efficiency between the same temperature limits.

It was assumed that the minimum temperature to which the exhaust gases can be cooled, with reasonable sizes of heat exchanger for the bottoming cycle, is about 100°C (370°K). The bottoming cycle then converts the excess energy in the exhaust gases between the stack temperature and the exit temperature, with the assumed effectiveness. These assumed values of effectiveness are reasonable and well within the current state of art for these engines [1,9].

(d) It was found that in the computer simulation of the LHR version of the engine, the convective heat loss would be reduced to approximately half its value for the water-cooled engine, in all throttle notch positions, when a value of 1000°K was assumed as the average temperature for the charge confining surfaces. This probably represents the upper limit from the point of view of reliability and endurance, particularly for a locomotive engine where long durations between overhauls are demanded.

It was also found that about 30 percent of the energy thus retained was converted directly to work. This agrees with other recent findings [11] corroborating the general soundness of the heat transfer model employed.

C.1.4 Results of the Computer Simulation

Some of the results of the computer simulation of the two versions of the engine – the conventional water-cooled version and the LHR version – are included as Annexes A and B to this appendix. The various values output refer to each cylinder of the 16-cylinder engine.

At each throttle notch position, the amount of fuel injected was adjusted to obtain the output specified in Table C.2 for that notch position in the case of the conventional water cooled engine. The same amount of fuel produces a higher output at the same notch position in the LHR version. This has been taken into account in compiling Table C.3, which presents the total fuel consumption incurred in 100 hours of operation on the heavy duty cycle. In this table, it has been assumed that the various modifications and additions have no effect on fuel consumption during the periods of idling and dynamic braking.

Table C.4 shows the additional power provided by the bottoming cycle engine in each case, together with the fuel savings that can be achieved in 100 hours of operation over the duty cycle, assuming that the same power is demanded from the powerplant in every case.

The fuel savings possible in litres is based on the assumption that the specific gravity of the fuel is 0.9, which is a representative figure for diesel fuels.

TABLE C.3

FUEL CONSUMPTION IN 100 h OPERATION ON HEAVY-DUTY CYCLE

		Conventional - Water-Cooled						LHR					
		TC		TC & Rankine		TC & Stirling		TC		TC & Rankine		TC & Stirling	
Throttle Position	Cycle of Time	Fuel Rate (kg/h)	Fuel Cons. (kg)	Fuel Rate (kg/h)	Fuel Cons. (kg)	Fuel Rate (kg/h)	Fuel Cons. (kg)	Fuel Rate (kg/h)	Fuel Cons. (kg)	Fuel Rate (kg/h)	Fuel Cons. (kg)	Fuel Rate (kg/h)	Fuel Cons. (kg)
Notch 1	3	55.6	166.8	47.6	142.8	44.3	132.9	46.9	140.7	39.0	117.	36.1	108.3
Notch 2	3	99.	297.	87.9	263.7	83.3	250.	91.6	274.8	78.0	234.	72.5	217.5
Notch 3	3	172.3	516.9	152.7	458.1	144.2	432.6	156.4	469.2	135.7	407.1	127.2	381.6
Notch 4	3	239.8	719.4	215.	645.	205.	615.	218.7	656.1	191.7	575.1	181.	543.
Notch 5	3	340.2	1020.6	304.7	914.1	290.	870.	312.	936.	274.	822.	259.5	778.5
Notch 6	3	426.7	1280.1	386.3	1158.9	368.8	1106.4	391.8	1175.4	347.5	1042.5	328.7	986.1
Notch 7	3	541.5	1624.5	491.	1473.	469.	1407.	499.	1497.	444.	1332.	420.	1260.
Notch 8	30	638.7	19161.	584.4	17532.	560.5	16815.	585.	17550.	527.	15810.	500.	15000.
Idle	41	26.	1066.	26.	1066.	26.	1066.	26	1066.	26.	1066.	26.	1066.
Dynamic Braking	8	26.	208.	26.	208.	26.	208.	26.	208.	26.	208.	26.	208.
Total Fuel Consumption (kg/100 h)			26060.		23862.		22903.		23973.		21614.		20549.

TABLE C.4

CONVENTIONAL AND LHR ENGINES COMPARED
Baseline Output from Conventional Engine: 2640 kW
100 Hours Operation on Heavy-Duty Cycle

	Conventional			LHR		
	Bottoming Cycle Engine Output (kW)	Fuel Savings (%)	Fuel Savings (litres/) (100 hrs)	Engine Output (kW)	Cycle Fuel Savings (%)	Bottoming Fuel Savings (litres/) (100 hrs)
TC	-	-	-	-	8	1878
TC & Rankine	240	8.4	1978	328	17	4001
TC & Stirling	368	12.1	2841	493	21.1	4960

It is difficult to analyze the cost effectiveness of the bottoming cycle systems. The cost of these engines depends on the volume of production. It was difficult to estimate the cost from the data given in reference 1 as there are some inconsistencies in that data. Therefore, if we assume tentatively that the bottoming cycle engine has the same cost per unit output as the main diesel engine (approx. $200.00 per kW) we can estimate the period in which the bottoming cycle engines pay for themselves. The cost of diesel fuel is also a highly variable and unpredictable parameter. Reference number 1 estimates that the cost of diesel oil will be around $0.35 (Canadian) per litre in bulk purchase. On this basis, both the Rankine and Stirling bottoming cycle engine systems pay back the initial capital costs in about 5000 hours of operation of the locomotive. Subsequently, the fuel savings afforded by the Stirling engine system will be higher than those of the Rankine engine system. These statements apply to both the conventional and the LHR versions of the engine. In the long run, the fuel savings with the LHR engine will exceed those associated with the conventional engine. On the other hand, one must not lose sight of the immense problems facing the successful development of the LHR engine before it can replace the well-proven, reliable water-cooled engine.

C.1.5 Conclusions and Recommendations

The fuel savings that can be achieved by recovering part of the exhaust energy with the help of bottoming cycles are substantial, particularly for the LHR engine. The outputs of both the Rankine and the Stirling versions of the bottoming cycle engine are fairly sizeable; the values of "effectiveness" assumed for these engines are therefore attainable within the current state of the art. In fact, it may even be possible to attain the assumed effectiveness of 30 percent for the Stirling engine with air as the working medium [12], eliminating the need to handle either hydrogen or helium. Walker [12] also suggests a simple way of converting a conventional reciprocating engine to a Ringbom-Stirling engine. It is quite easy to arrange that a few extra cylinders on the engine crankshaft execute the bottoming cycle, thus eliminating the power transmission problem.

The LHR version of the engine offers a substantial improvement in fuel savings. However, a considerable amount of development work is needed before a successful version of the LHR engine is produced. The material and tribological problems in the LHR engine are indeed formidable and pose a threat to the reliability which is essential for railroad operation. It is noteworthy that, despite more than a decade of intensive effort, there is not a single prototype of the LHR engine ready for large-scale manufacture.

In addition to the Rankine and Stirling engines, there are other forms of heat engines that may be investigated for use in the bottoming cycle [13].

The research and development activities in this field that can be pursued gainfully in Canada have to be tempered by the fact that there is only one independent manufacturer of (locomotive) diesel engines (Bombardier). Engine research is carried out on a small scale in a number of academic institutions. Under these circumstances, it may be too ambitious to embark on the development of an LHR engine. Nevertheless, it is worthwhile to initiate work on incorporating bottoming cycles on conventional water-cooled engines.

C.2 EVOLUTIONARY DIESEL HARDWARE IMPROVEMENTS

C.2.1 Efficiency of Existing Locomotive Engines

When the JPL report was published in 1981, the conventional (no bottoming cycle) locomotive medium-speed diesel engine was considered highly efficient, with little potential for improvement. Nevertheless, small but significant decreases in full-load, brake-specific fuel consumption (BSFC) have since been achieved. Refinement of the existing General Motors Electro-Motive Diesel (EMD) and General Electric (GE) designs, rather than any revolutionary changes, have led to these improvements.

Although full-load BSFC is a good indication of the cycle efficiency of the powerplants, improvements in fuel consumption have also been made by reducing idle speeds, eliminating parasitic losses, etc.

The popular EMD model 645F3 engine achieved a 6.1 percent reduction in full-load BSFC between 1980 and 1983 [14]. An increase in compression ratio, improved injection, modified valve timing, increased turbocharger efficiency and "fire ring" relocation contributed to the improvement. The full-load BSFC of the four-stroke GE 7FDL-16 engine was improved to a lesser degree during the same time period, as this engine was already highly fuel efficient [14]. An improved aftercooler and exhaust system, higher turbocharger efficiency, and reduced piston weight and friction produced the improvements for the GE engine.

Commonly used engines manufactured by EMD, GE and ALCO/Bombardier were originally designed when fuel consumption was less important due to the low and stable oil prices of the time. Improvements since that time have reduced fuel consumption, but it might be expected that a new design specifically intended to provide better fuel economy would offer greater potential gains than refinements to old designs. The new Caterpillar 3600 series engines were designed from the beginning for minimum fuel consumption [15]. However, in spite of this design effort, its full-load BSFC does not appear to be greatly superior to the refined versions of the older engines. The full-load BSFC figures are as follows:

EMD - 0.336
GE - 0.328
Caterpillar - 0.327

It can be seen that the BSFC of the Caterpillar unit differs little from the improved GE four-stroke. This suggests that combustion efficiency with conventional engines may have peaked. Improvement trends at the level shown by EMD are therefore unlikely to continue through the type of refinements previously used. On the other hand, fuel consumption may improve if the efficiency can be maintained closer to the full-load figure throughout the operating range.

C.2.2 Fuel Injection

C.2.2.1 Introduction

Advances in fuel injection equipment design have contributed significantly to the improved full-load BSFC of existing medium-speed diesel engines using conventional fuel. However, fixed injection timing optimized for full load, and the decrease in injection pressure which accompanies reduced engine speed, hamper the achievement of the potential combustion efficiency throughout the rest of the load and speed range. Better control of the fuel injection process, and hence of combustion, will be required if peak cylinder pressures are to be reduced or maintained while ignition delay (due to lower cetane fuel) and brake mean effective pressure (BMEP) (due to demand for increased power) increase.

In response to these needs, activities in fuel injection research have increased in the 1980s. Developments in fuel injection equipment can potentially have a greater impact on rail system fuel consumption than design improvements in new engines, since injection equipment could be retrofitted to existing engines. It has been suggested [19] that open-loop control of injection timing could reduce fuel consumption by up to about 5 percent. Closed-loop control would improve upon this somewhat, but its main advantage would be in desensitizing the engine to fuel composition and ambient conditions.

C.2.2.2 Injection Pressure Requirements

Maximum fuel line pressures have increased from the range 500-600 bar to 1100-1200 bar in the past twenty years. Injection pressures up to 1500 bar have been studied [16] and will likely rise further.

Any increases in peak cylinder pressure will require increased injection pressure to maintain the pressure drop across the nozzle, and hence atomization quality. If the viscosity of the fuel reaching the injector is allowed to increase, increased injection pressure will also be required [16].

As injection pressure increases, the injection period decreases, thus shortening the heat release time. This improves combustion efficiency [16,17,18], albeit at the expense of increased peak cylinder pressure. It is also important that increases in injection pressure be combined with accurately closing delivery valves [16,18].

The most significant injection pressure effect in present systems is the drop in injection pressure which occurs at lower engine speeds. Unit injectors are superior to pump-line-nozzle systems in this regard, and are considered better overall for high pressures [19,20].

General Motors EMD engines were formerly the only common locomotive powerplants to use unit injectors [17], but the recently introduced 3600 series engines from Caterpillar use a new unit injector of their own design [21].

C.2.2.3 Staged Injection

A staged injection system can reduce the ignition delay period considerably over a conventional system. The small pilot quantity of fuel injected early in the compression stroke undergoes oxidation reactions, and ensures rapid ignition of the main charge at near top dead centre. This lowers the peak cylinder pressure at a given BMEP, particularly with lower cetane fuels.

Further development of the staged injection concept is required, but it has been successfully demonstrated experimentally using two injectors per cylinder [22], and in practice using a single injector [23].

Optimum timing and quantity for the pilot and main charges must be determined for the engine and fuel in question. Implementation may ultimately be accomplished most satisfactorily through electronic injection controls.

C.2.2.4 Electronic Injection

For maximum overall efficiency, injection timing must vary throughout the load and speed range. If staged injection is implemented, then this adds further control requirements. These demands can be met using electronically controlled spill valves in the fuel injection circuit, termed electronic fuel injection. Single injection stage systems have already been tested [24,25,35]. The Detroit Diesel Allison (DDA) electronically controlled unit injector has been demonstrated in practice [35] and may soon be applied to production engines [26].

The control signals needed by such a system are presently open loop; i.e. based upon an experimentally determined engine map. Bigger improvements could be produced if a closed-loop controller could be developed [19]. Ideally, immediate cylinder pressure data would be used as an input for such a system. The reliability and cost of present cylinder pressure transducers make this difficult. However, it has been demonstrated that accurate indicator diagrams can be obtained without such transducers through measurements of the strain history of cylinder head bolts [27]. Such a closed-loop system has the potential to maintain optimum injection conditions regardless of fuel composition and ambient conditions.

There is also work in progress on using crankshaft speed variations as a combustion diagnostic tool [28], but this is less promising as a closed-loop feedback control variable.

C.2.3 Conclusions

It is clear that improvements in performance and reliability can be gained by the incorporation of sophisticated electronic controls in diesel engines. The accelerating technology of such controls in automotive engineering can be adapted to locomotive requirements. Open-loop electronic fuel injection will likely appear on production high-speed engines before 1990 [26]. The incentive to develop closed-loop systems does not yet appear to exist. Closed-loop controllers will not likely become attractive until fuel quality declines enough to require such systems in order to maintain present output and efficiency. It would be prudent to start development before a shortage of high-quality fuel occurs.

Canada's limited diesel manufacturing capability makes it difficult to conduct major development programs for complete engines. On the other hand, Canadian electronic firms are among the world leaders in applications of the new technology. It should therefore be possible for

Canada to play an active role in developing the new electronic control systems and subcomponents needed for future medium-speed diesels.

C.3. FUEL CONSIDERATIONS

C.3.1 Low-Grade Hydrocarbon Fuels

C.3.1.1 Introduction

Lower grade fuels derived from petroleum, tar sands, coal or shale can have ignition properties closer to conventional diesel than methanol or natural gas. The term "low grade" correctly implies that characteristics important to diesel operation (cetane number, viscosity, wear-accelerating impurities) become less favourable as fuel grade and price become lower. However, the complexity of the blend of compounds, elements, and metals in such fuels can give rise to innumerable permutations of these critical properties. Thus, for instance, one low-viscosity fuel may have a lower cetane number and more impurities than another very high viscosity fuel. Any definition of fuel quality must encompass all the pertinent fuel properties.

EMD [29] has recommended a set of essential fuel property limits for "reasonable component life." Poor quality marine diesel fuel and low-viscosity intermediate fuel oil (if heated) would fall within these guidelines. EMD predicts overhaul periods approaching one-third those with quality gas oil with such fuels.

C.3.1.2 Viscosity

With petroleum-derived fuels, viscosity generally increases as price decreases. The fuel is difficult to pump and atomize, and combustion efficiency suffers. EMD recommends a maximum viscosity of 55 SUS (Saybolt Universal Seconds) at 100°F. Jacket water heating can permit an increase to 150 SUS, while additional heating to 195°F allows a further increase to about 200 SUS. Medium-speed marine diesels have been reported to operate with fuel viscosities as high as 700 cSt (approx. 3200 SUS) with a typical fuel inlet temperature of 147°C [30,31,32,33].

Fuel heating requires additional hardware, and waste heat from coolant and/or exhaust would not be available during cold starting. Winter temperatures would lead to further problems with tanks and fuel lines. Nevertheless, marine diesel experience indicates that very high fuel viscosity alone is not a fundamental limitation to successful engine operation. Fuel viscosity limits would be set by the ability to heat the fuel reliably and economically.

C.3.1.3 Cetane Number

EMD recommendations allow a drop in cetane number to 37 [2]. Tar sand fuels with a cetane number of as low as 30 have been used successfully by CN Rail in Canada [34]. Ignition delay increases as cetane number decreases, until knock becomes unacceptable. It is the ignition delay period, rather than the auto-ignition temperature of the fuel, that limits reductions in cetane number.

DDA research has shown that the cylinder air temperature required before injection in order to achieve ignition rises very little in the high to mid cetane range - less than 50°F down to a cetane number of 15 [35]. Thus, fuels with cetane numbers in this range could be used if knocking could be eliminated.

As discussed earlier, one method of achieving this is staged fuel injection [35,36,22,23]. Ignition delay is shortened by injecting a small pilot quantity of fuel early in the compression stroke. This fuel undergoes oxidation reactions before the main charge is injected near top dead centre. Knock-free operation with 17.7 cetane fuel has been demonstrated with a two-cylinder EMD research engine using staged injection [22]. A simple form of pre-injection system has been tested in service in a French Navy vessel [23]. DDA predicts that electronically controlled single-stage injection, along with glow plugs for starting and light loads, could allow operation with fuels in the 20-60 cetane range [35].

Precise control of timing and quantity for the pilot and main charges is essential. This task is best undertaken by some form of electronically controlled fuel injection. The evolution of electronic fuel injection aimed at reducing the consumption of conventional diesel fuel (particularly at light loads) should, as a consequence, permit the flexibility needed for reducing cetane requirements in the future.

C.3.1.4 Impurities

Even if a fuel can be injected and burned efficiently, excessive wear due to impurities can eliminate any savings in fuel cost. Minute quantities (less than 100 ppm) of catalyst fines and vanadium can lead to component failure. Sulphur and carbon residue should also be limited. Levels of harmful impurities tend to be higher in the higher-viscosity petroleum-derived fuels. In these cases, impurity levels, rather than heating requirements to reduce viscosity, could limit usage in existing engines.

Many newer medium-speed marine diesel engines are capable of tolerating the very high impurity levels of heavy fuel oil (Bunker C) [30,31,32,33]. These engines have been designed or re-designed for that purpose, and include the use of materials resistant to corrosive and abrasive attack. Conversion of existing engines to use low-grade fuel blends is feasible, but converted engines can demand three to ten times more maintenance time than those designed specifically for low-grade fuels [37]. The latter are designed so that not only is less maintenance required, but also the maintenance needed is easier and faster. This implies that the maintenance requirements of such new engines will decrease even further if a high-grade fuel is used.

The new Caterpillar medium-speed engines are designed to cope with anticipated increases in fuel impurity levels [21]. Many special features are made available as options to allow operation on blended or residual fuels. The following lists show additional items or basic design features provided to address three of the principal concerns:

Viscosity	Sulphur	Vanadium
Fuel heaters	Increased liner temperature	High overlap camshaft
Fuel centrifuges	Threaded valve guides	Water-cooled valve seats
Auxiliary filters	Injector tip cooling	Valve rotators
Auxiliary fuel pumps	Increased jacket water temperature	Reduced ratings
	Water temperature controls	Nickel-chrome valves
	Increased oil consumption	
	Increased oil TBN	

A combination of these features can be selected, as required by the specific characteristics of the fuel to be used. If auxiliary fuel-handling devices (listed above under viscosity) and derating were required, power-to-weight ratio would suffer.

C.3.2 Natural Gas

Natural-gas-fuelled compression ignition engines must be dual-fuel engines, as the injection of a high-cetane fuel is required to initiate combustion. Converting the powerplant to a spark ignition engine allows operation with natural gas only, but the potential efficiency of the dual-fuel engine is higher. Pilot injection of diesel provides a multiple source ignition superior to spark systems, thus allowing leaner operation before throttling is required [38,39]. When this limit is reached, the dual-fuel engine remains unthrottled and reverts to pure diesel operation.

Staged injection (SI) gas engines which exploit lean burn concepts developed originally for automotive engines have improved lean limits. Although they still do not allow running on as lean a gas mixture as engines with diesel pilot injection, the difference is reduced. This has been demonstrated in tests on the new Bombardier 251 spark ignition engine [40].

The dual-fuel compression ignition engine has evolved primarily for stationary engine applications. LNG carrier ships are another application of this engine type [41].

EMD produces a dual-fuel version of their Model 567 in naturally aspirated form [42,43]. ALCO discontinued dual-fuel engine development in the early 1960s but Bombardier is now developing a turbocharged dual-fuel engine suitable for locomotive applications. The design goal is that this engine have equal power and reliability in either diesel or gas mode [40].

In highly turbocharged applications new methods of gas admission to the cylinder will be required if similar power is to be produced. The present systems admit the natural gas by means of a separate valve either directly into the cylinder (EMD) or just outside the air inlet port (Bombardier). Both approaches result in fuel being present in the cylinder during the compression stroke and hence the possibility of knock. Bombardier is considering direct injection of LNG as a solution. Even with equivalent engine power, the power/weight ratio of the engine-fuel system as a whole would suffer, as the volumetric storage requirements for CNG and LNG are approximately 6.5 and 1.7 times that of diesel, respectively [44].

As noted above, present dual-fuel engines must be operated in diesel mode at idle and light loads, since the gas-air mixture becomes too lean to be burned efficiently. However, deliberate differences in gas delivery to different cylinders could allow idle and low power operation on gas. This might be achieved by running one or more cylinders at high enough loads to allow efficient operation on gas while at the same time stopping fuel delivery altogether to the others. Overall engine power required would thereby be obtained without excessive leaning of any running cylinder. Electronic control to allow different cylinders to be fuelled in rotation would allow the entire engine to be kept at operational temperature. The irregular crankshaft loads which such a system would entail might pose difficulties, but in principle the concept would remove the present combustion barrier to full load-range fuelling (including idle) on natural gas.

Another possibility to reduce the diesel fuel requirement would be to eliminate most of the idling time fuel waste by adding a natural-gas-fuelled heating system so as to maintain a satisfactory coolant temperature and allow engine shut-down. Such a heater is much simpler, more reliable and requires less maintenance than a similar device burning diesel fuel. It is therefore more practical for operation on natural gas.

Natural gas leaves fewer deposits than diesel fuel. It should therefore be possible to prolong the maintenance interval of the cylinder head, liner and piston [41]. The practical effect of this has not yet been quantified.

Canadian research activities in dual-fuel natural gas engines presently include work at Bombardier in Montreal, the University of Calgary, and the University of British Columbia. Experimental work formerly conducted at the National Research Council Canada (NRCC) on natural gas in a medium-speed diesel was completed some time ago and the engine was returned to its owners (a mining operator).

C.3.3 Methanol

Converting natural gas into methanol provides a fuel which is liquid at ambient temperatures, thus simplifying the fuel storage, transportation, and handling requirements. As a liquid, it can be injected directly into the cylinder in the same manner as diesel fuel. This method of fuel admission avoids the knock limits imposed by homogeneous charges in highly turbocharged applications.

Methanol has a cetane number of less than 5 and therefore has an excessive ignition delay for a conventional diesel engine. Nevertheless, methanol will autoignite provided that a high enough temperature exists in the cylinder at the time of injection.

At the time the JPL report was published (1981), the literature was generally negative towards methanol in compression ignition (CI) engine applications, and most demonstrations involved diesel pilot injection for ignition. Since that time, Detroit Diesel Allison has pioneered major breakthroughs in CI methanol engine technology [45,46].

In 1983-84 DDA successfully demonstrated a pure-methanol-fuelled transit coach in California. The engine was a retrofitted version of their production 6V-92TA (turbocharged) two-stroke diesel. In this engine, a cylinder temperature sufficient to ignite methanol is achieved by lowering the scavenge ratio. This allows a higher level of hot residuals to remain in the cylinder. The scavenge blower and turbocharger combination matches the air flow requirements for most of the load and speed range. A bypass valve in the scavenge blower provides additional scavenge ratio control. Special electronic unit injectors allow flexibility in injection timing and provide the higher fuel flow rates required with methanol. Glow plugs are used for starting and light loads. This engine produces the same output as the diesel version. In its present state of development, thermal efficiency is equal or slightly better than the diesel-fuelled version throughout the load range. Further improvements in thermal efficiency are expected. Part of the advantage of the methanol engine is due to its electronic injection system, which could also improve performance in the conventionally fuelled engine.

DDA has also proposed a multi-fuel version of their 6V-53T engine for the U.S. Army Tank Automotive Command (TACOM) [35]. The modifications would be similar to the existing coach engine, and operation on fuels with cetane numbers from 60 down to as low as 0 would be possible. Fuel heating systems would allow high-viscosity fuels to be used as well. DDA estimates approximately three years for the development of this system.

The unique characteristics of the two-stroke engine allow relatively easy control of in-cylinder air temperatures compared to four-stroke engines. DDA has also analytically investigated multi-fuel concepts with their 6.2 litre indirect injection four-stroke diesel. They predict the possibility of operation on neat methanol, but the necessary control systems and technology are not yet available. Cummins has also proposed a program to adapt their four-stroke direct injection engine for neat methanol operation. Both the DDA and Cummins programs have been stimulated by proposed heavy-duty vehicle particulate and NO_x emission regulations.

In comparison with dual-fuel natural gas diesels, the neat-methanol-fuelled two-stroke compression ignition engine is new and of unproven reliability. It has yet to be demonstrated in the case of the four-stroke. However, DDA's experience to date with their two-stroke engine using existing technology has not revealed any insurmountable problems. Single-fuel operation throughout the load and speed range is possible, and the power output matches that of the conventionally fuelled diesel engine. The lower flame temperature of methanol may in fact allow higher BMEP than diesel fuel if the output of the engine in question is limited by thermal considerations rather than purely mechanical factors.

DDA's two-stroke temperature control system lends itself well to multifuel concepts. Thus, the transition from diesel, through lower cetane hydrocarbon fuels, and finally to methanol could be accomplished using essentially the same engine hardware. Retrofits to existing engines might also be accommodated.

The mass of methanol required to match the energy content of diesel fuel is about 2.15 times that of the diesel. The densities are similar (0.796 g/cc methanol, 0.817 DF2 at 60°F).

The total mass of methanol fuel plus its container is much less than that of CNG, but probably more than that of LNG depending upon the cryogenic storage container weight.

C.3.4 Conclusions

Utilization of low-grade and alternative fuels is an important long-term goal. Although the present economic conditions have resulted in a large decrease in oil prices, this is at best a temporary respite. It is very much in the interest of the Canadian railroads that these fuel options be pursued while time still remains.

The efficient combustion of even the worst foreseeable hydrocarbon fuels, whether derived from petroleum, tar sand, coal, shale or other, poses no insurmountable technical barriers. However, the cost and complication of retrofitting hardware to reduce fuel viscosity (if necessary) and improve injection control (eliminate knock) may prove excessive. Engine durability in the presence of impurities (beyond present limits) is possible only through re-design.

Solutions to the wear, corrosion, special handling and other problems related to low-quality fuels can best be undertaken by collaborative efforts between manufacturers and large research institutions. In Canada, the National Research Council is best able to undertake the sort of lengthy durability trials needed for particular fuels. In view of the Canadian interest in tar-sands-derived fuels, as opposed to the oil shale products of most interest to the USA, ongoing effort should be supported and encouraged in this area.

Given the success to date in the use of natural gas, and the projected availability of this fuel in Canada, continued research on its applications is appropriate. In this area, the Canadian universities and small research organizations already involved in engine performance studies can make a valuable contribution at modest cost. It is reasonable to take all steps necessary to ensure the technology is developed and refined which will allow natural gas use in rail locomotion in Canada.

Because of the increasing availability of methanol produced in Canada from natural gas, this fuel also merits further attention for railway use. It has a number of advantages compared to direct firing of natural gas. Again, research is warranted to develop the technology so that it will be available at known cost when needed. Methanol research is actively under way in Canada for automotive applications and the extension to medium-speed diesel operation should be encouraged.

C.4. GAS TURBINES FOR RAILWAY PROPULSION

Turbine engines for railway locomotion were first used in the early 1930s [1]. Installations to date may be categorized into two classes directly related to the origin of the turbine. These are: industrial ground installation type, and aircraft turboprop type gas turbines. The former are generally large units, bulky, heavy and of high power rating. They have not been particularly efficient as gas turbines go, their high power output has been difficult to distribute and they usually end up operating for the majority of their life at part load, at which they are even less efficient.

Adapted aircraft turbine technology has been more successful in past rail applications. Union Pacific employed fifty-five turbine locomotives of up to 10,000 hp rating in the 1950s [1] until residual oil costs rose due to improvements in oil refining methods that allowed economic residual oil conversion into more saleable middle distillates.

In the 1960s and 1970s United Aircraft Corporation and General Motors Electro-Motive Division developed, demonstrated and displayed spin-off air industry gas turbines as individual or grouped power sources for rail locomotion. However, continuing improvements in diesel performance occurring at the same time and the large increases in liquid fuel costs that were symptomatic of the 1970s, discouraged turbine train applications due to their higher fuel consumption.

Turbine engines are based on the Brayton Cycle. As a consequence, regeneration is practical and can greatly improve efficiencies, especially at part-load conditions. The recent advances in ceramic regenerators developed for automotive applications have made this possibility much more attractive. For comparable efficiencies, unregenerated gas turbines (GTs) need very high pressure ratios, whereas regenerated cycles give best fuel economy at quite low pressure ratios. Low pressure ratios offer substantial cost savings.

Turbine inlet temperatures in aviation gas turbines continue to climb, thereby increasing efficiencies. Types I and II ceramics permit temperatures to 3000°F. Hoppenheimer [54] points out that the carbon-sintered composites may soon yield permissible inlet temperatures of 3500°F. Ling Tempco Vought (LTV) has already manufactured a 15-inch turbine wheel of 36 blades that withstood this temperature while operating at 40,000 rpm. At 7.5 lbs, its weight is only 25 percent of that of an equivalent superalloy part that could only operate at 2400°F.

As a longer-term development, Epskin [55] draws attention to the possibility of closed-loop feedback control on turbine engine performance, specifically in terms of increasing on-and-off-design performance. Additionally, such active closed-loop control could reduce downtime by making the engine more mission-adaptive. He also holds out great hope for significantly reducing friction, again contributing to improved fuel economy.

Fuel-efficient shaft turbines are under intensive development for automobiles. Turbines already available on the market for trucks and other heavy road vehicles give values of BSFC around 0.40-0.45 lb/hp-h. This is still some 20-30 percent more than medium-speed diesels. However, current experimental regenerated automotive gas turbines such as the AGT 101 are projected to achieve 0.35 with turbine inlet temperatures of 2100°F and below 0.30 lb/hp-h for 2500°F.

Since there is active development of both vehicular- and aircraft-derived gas turbines, the question arises as to which, if either, of these industries is likely to produce a shaft turbine which can compete successfully with the medium-speed diesel in railroad use.

There is a fundamental difference in the cycles in these two applications in that aircraft gas turbines are unregenerated, high-pressure ratio devices, whereas automotive engines must be regenerated low-pressure machines in order to attain both competitive fuel economy and reasonable initial cost. For a given turbine inlet temperature, defined by similar materials considerations in both cases, an unregenerated GT has an optimum pressure ratio which is much higher than that of a regenerated cycle. At the peak turbine inlet temperatures of current metal components, an unregenerated turbine engine would have a pressure ratio in the range of 25-30, whereas its regenerated vehicular counterpart would have a pressure ratio around 5.

The gas turbines which will become available for railway use as offshoots of the other two industries will therefore be different. Automotive-derived machines will have low pressure ratios, regenerators, lower initial costs and probably lower maintenance costs once the technology is well established and in production. Aircraft-derived machines will have high pressure ratios, and probably higher initial costs, but offer well known total life-cycle costs, longer operational life, maintenance modularity and the great advantage of well-established technology. The fact that costs, maintenance considerations, personnel training requirements, etc., are well defined in the case of aircraft-derived gas turbines makes them more attractive for railway applications than their inherent cycle characteristics would suggest. Ongoing aviation developments and the potential large-scale implementation of gas turbines for automobiles offer promising possibilities for railway use at minimum cost. Wilson [9] points out that engine costs are proportional to the unit size and production rate and that production costs for millions of parts produced per year tend to approach 125 percent of material costs. Future non-metallic gas turbine engines may therefore have considerable cost advantages.

Gas turbines possess many attractive features for application as locomotives. The small, multiple unit system requires engines of the size range already designed and produced in Canada. The market for locomotive gas turbines should be available to Canadian industry when the time comes.

For the present, developments of fuel-efficient shaft turbines for automobile and truck applications should be monitored. Active support for these research and development activities by Canadian railroad interests is not recommended at present.

We do not consider closed-cycle gas turbines to be likely contenders for rail propulsion in the near or medium term. Although they have some potential advantages, their complexity is a serious deterrent to mobile applications. Even in stationary power generation applications, they have made only slight headway world-wide against open cycle plants.

C.5. SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

C.5.1 Comments on the JPL Report

Generally, the predictions of the JPL report are still valid in principle after five years. However, the new technology in various areas now appears likely to reach the production and practical utilization stage somewhat later than anticipated. For example, the report predicted that coal-fired steam engines would enter railway use in the late 1980s, whereas that now seems unlikely. As is often the case, the rendering of new technology into practice takes longer than anticipated. The removal of the goad to action previously supplied by high world oil prices has contributed to retarded development in alternative fuels areas.

The recommendation that work proceed on adiabatic diesel technology is still justified. However, in our opinion, the economy gains to be expected are somewhat smaller than the predictions of the JPL report and proponents of the scheme. The cycle calculations upon which the JPL estimates were made were rather unsophisticated and omitted certain important aspects. Among these are radiative heat transfer and the fact that an adiabatic engine implies no net heat transfer out of the block but does not mean no heat transfer between charge and cylinder walls. The heat transfer rate to the walls will probably not be greatly changed by the insulating refractory materials used. There is, therefore, a heat loss to the walls during the high temperature parts of the cycle, and a gain from the walls during the cooler parts. This heat gain during the compression process increases compression work and could have a significant effect upon cycle efficiency. There is a similar, although smaller, beneficial effect from heat addition during expansion. Cycle models which simply set heat transfer to zero omit this potentially important consideration. Further modelling effort is therefore recommended.

C.5.2 Specific Research Recommendations

It is recommended that the following specific research studies be conducted:

(1) A detailed study of the current state-of-the-art in the two most promising bottoming cycles which might be used to increase fuel economy while maintaining the medium-speed diesel as the main propulsion unit. These are Rankine and Stirling cycles. It is not known whether there is Canadian expertise in the former, but Stirling technology has been under active development at one Canadian university for many years. The current performance and costs of these devices could be used to estimate the fuel economy gains and the likely costs and operational implications.

(2) A study of electronic control of fuel injection and of other cycle controls such as running with various cylinders at different fuelling rates as discussed in Section C.2.2. This study should determine whether specific Canadian development projects are warranted for particular hardware applications.

(3) A study on the use of plasma torch and similar high-energy ignition systems as possible means for increasing natural gas and methanol performance in medium-speed diesels.

(4) Experimental work on staged ignition. This has the potential to greatly affect engine performance on a wide variety of fuels, including reduced-quality tar-sands-derived Canadian diesel. Fundamental studies in single cylinder research engines are recommended.

(5) Tests to evaluate the effects of impurities on engine components and determine appropriate material and other solutions.

(6) Continued work on natural gas performance generally, and in particular, means of improving the low load flammability limit and suppressing knock.

(7) Studies related to the use of methanol in medium-speed diesels. In particular, the influence of cylinder size on the encouraging results found by DDA in their multifuel, two-stroke, automotive-size engines.

(8) A study of closed-loop injection control using cylinder pressure data or other appropriate feedback.

(9) A study of natural gas injection directly into the cylinder as a means of avoiding knock in highly turbocharged operation.

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ANNEX A

PERFORMANCE OF THE CONVENTIONAL ENGINE WITH AND WITHOUT BOTTOMING CYCLES AT DIFFERENT THROTTLE NOTCH POSITIONS

NOTCH 8

```
                              OPERATING CONDITIONS

                                   R P M =   1097.00
                       COMPRESSION RATIO =     11.50
               INJECTION ADVANCE,DEGREES =     20.00
                   INLET PRESSURE (BARS) =      3.53
                   INLET TEMPERATURE (K) =    401.50
                  IGNITION DELAY,DEGREES =      7.00
                    HEAT TRANSFER OPTION =    1

                              MODEL PREDICTIONS

                      INDICATED WORK (J) =     .19576E+05
             HEAT LOST BY CONVECTION (J) =     .11273E+05
                       GAS RADIATION (J) =     .16380E+03
                      SOOT RADIATION (J) =     .56272E+03
              HEAT LOST BY RADIATION (J) =     .72651E+03
                  IND THERMAL EFFICIENCY =     .36580E+00
            IND OUTPUT PER CYLINDER (KW) =     .17895E+03
                            I M E P (PSI)=     .25938E+03
                            I M E P (BAR)=     .17883E+02
                            F M E P (PSI)=     .19508E+02
                            B M E P (PSI)=     .23987E+03
            BRAKE OUTPUT PER CYLINDER,kW =     .16550E+03
           BRAKE OUTPUT PER CYLINDER,BHP =     .22184E+03
                 BRAKE THERMAL EFFICIENCY=     .33829E+00
                         B S F C, g/kW.h =     .24121E+03
             FUEL CONSUMPTION RATE, kg/h =     .39920E+02
            EXHAUST STACK TEMPERATURE, K =     .75148E+03

          BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=     .15358E+02
         ADDITIONAL OUTPUT - STIRLING, kW=     .23037E+02
 RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =     .36968E+00
 STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=     .38538E+00
             FUEL SAVINGS - RANKINE,kg/h =     .33900E+01
            FUEL SAVINGS - STIRLING,kg/h =     .48778E+01
```

NOTCH 7

```
                              OPERATING CONDITIONS

                                   R P M =    987.00
                       COMPRESSION RATIO =     11.50
               INJECTION ADVANCE,DEGREES =     20.00
                   INLET PRESSURE (BARS) =      3.18
                   INLET TEMPERATURE (K) =    401.50
                  IGNITION DELAY,DEGREES =      7.00
                    HEAT TRANSFER OPTION =    1

                              MODEL PREDICTIONS

                      INDICATED WORK (J) =     .18295E+05
             HEAT LOST BY CONVECTION (J) =     .10463E+05
                       GAS RADIATION (J) =     .19169E+03
                      SOOT RADIATION (J) =     .66137E+03
              HEAT LOST BY RADIATION (J) =     .85305E+03
                  IND THERMAL EFFICIENCY =     .36281E+00
            IND OUTPUT PER CYLINDER (KW) =     .15048E+03
                            I M E P (PSI)=     .24242E+03
                            I M E P (BAR)=     .16714E+02
                            F M E P (PSI)=     .21014E+02
                            B M E P (PSI)=     .22140E+03
            BRAKE OUTPUT PER CYLINDER,kW =     .13743E+03
           BRAKE OUTPUT PER CYLINDER,BHP =     .18423E+03
                 BRAKE THERMAL EFFICIENCY=     .33136E+00
                         B S F C, g/kW.h =     .24626E+03
             FUEL CONSUMPTION RATE, kg/h =     .33844E+02
            EXHAUST STACK TEMPERATURE, K =     .78371E+03

          BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=     .14153E+02
         ADDITIONAL OUTPUT - STIRLING, kW=     .21229E+02
 RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =     .36548E+00
 STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=     .38254E+00
             FUEL SAVINGS - RANKINE,kg/h =     .31598E+01
            FUEL SAVINGS - STIRLING,kg/h =     .45283E+01
```

NOTCH 6

```
                          OPERATING CONDITIONS

                                  R P M =    889.00
                      COMPRESSION RATIO =     11.50
              INJECTION ADVANCE,DEGREES =     20.00
                  INLET PRESSURE (BARS) =      2.86
                  INLET TEMPERATURE (K) =    401.50
                 IGNITION DELAY,DEGREES =      7.00
                   HEAT TRANSFER OPTION =    1

                           MODEL PREDICTIONS

                     INDICATED WORK (J) =       .16014E+05
            HEAT LOST BY CONVECTION (J) =       .93523E+04
                      GAS RADIATION (J) =       .19013E+03
                     SOOT RADIATION (J) =       .68626E+03
             HEAT LOST BY RADIATION (J) =       .87639E+03
                 IND THERMAL EFFICIENCY =       .36299E+00
           IND OUTPUT PER CYLINDER (KW) =       .11864E+03
                           I M E P (PSI)=       .21220E+03
                           I M E P (BAR)=       .14630E+02
                           F M E P (PSI)=       .21736E+02
                           B M E P (PSI)=       .19046E+03
           BRAKE OUTPUT PER CYLINDER,kW =       .10649E+03
          BRAKE OUTPUT PER CYLINDER,BHP =       .14275E+03
                BRAKE THERMAL EFFICIENCY=       .32581E+00
                        B S F C, g/kW.h =       .25045E+03
            FUEL CONSUMPTION RATE, kg/h =       .26670E+02
           EXHAUST STACK TEMPERATURE, K =       .77684E+03

         BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

         ADDITIONAL OUTPUT - RANKINE, kW=       .11116E+02
        ADDITIONAL OUTPUT - STIRLING, kW=       .16674E+02
RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =       .35982E+00
STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=       .37683E+00
            FUEL SAVINGS - RANKINE,kg/h =       .25209E+01
           FUEL SAVINGS - STIRLING,kg/h =       .36107E+01
```

NOTCH 5

```
                          OPERATING CONDITIONS

                                  R P M =    779.00
                      COMPRESSION RATIO =     11.50
              INJECTION ADVANCE,DEGREES =     20.00
                  INLET PRESSURE (BARS) =      2.50
                  INLET TEMPERATURE (K) =    401.50
                 IGNITION DELAY,DEGREES =      7.00
                   HEAT TRANSFER OPTION =    1

                           MODEL PREDICTIONS

                     INDICATED WORK (J) =       .14431E+05
            HEAT LOST BY CONVECTION (J) =       .84247E+04
                      GAS RADIATION (J) =       .22154E+03
                     SOOT RADIATION (J) =       .81432E+03
             HEAT LOST BY RADIATION (J) =       .10359E+04
                 IND THERMAL EFFICIENCY =       .35945E+00
           IND OUTPUT PER CYLINDER (KW) =       .93681E+02
                           I M E P (PSI)=       .19121E+03
                           I M E P (BAR)=       .13184E+02
                           F M E P (PSI)=       .23112E+02
                           B M E P (PSI)=       .16810E+03
           BRAKE OUTPUT PER CYLINDER,kW =       .82358E+02
          BRAKE OUTPUT PER CYLINDER,BHP =       .11040E+03
                BRAKE THERMAL EFFICIENCY=       .31601E+00
                        B S F C, g/kW.h =       .25822E+03
            FUEL CONSUMPTION RATE, kg/h =       .21267E+02
           EXHAUST STACK TEMPERATURE, K =       .80826E+03

         BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

         ADDITIONAL OUTPUT - RANKINE, kW=       .95665E+01
        ADDITIONAL OUTPUT - STIRLING, kW=       .14350E+02
RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =       .35271E+00
STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=       .37107E+00
            FUEL SAVINGS - RANKINE,kg/h =       .22132E+01
           FUEL SAVINGS - STIRLING,kg/h =       .31556E+01
```

NOTCH 4

```
                         OPERATING CONDITIONS

                                  R P M =      680.00
                      COMPRESSION RATIO =       11.50
             INJECTION ADVANCE,DEGREES =       20.00
                  INLET PRESSURE (BARS) =        2.20
               INLET TEMPERATURE (K) =      401.50
               IGNITION DELAY,DEGREES =        7.00
                 HEAT TRANSFER OPTION =      1

                          MODEL PREDICTIONS

                    INDICATED WORK (J) =         .11692E+05
         HEAT LOST BY CONVECTION (J) =         .71797E+04
                    GAS RADIATION (J) =         .19782E+03
                   SOOT RADIATION (J) =         .78839E+03
          HEAT LOST BY RADIATION (J) =         .98621E+03
              IND THERMAL EFFICIENCY =         .36056E+00
         IND OUTPUT PER CYLINDER (KW) =         .66253E+02
                       I M E P (PSI)=         .15492E+03
                       I M E P (BAR)=         .10681E+02
                       F M E P (PSI)=         .23364E+02
                       B M E P (PSI)=         .13155E+03
        BRAKE OUTPUT PER CYLINDER,kW =         .56261E+02
       BRAKE OUTPUT PER CYLINDER,BHP =         .75417E+02
              BRAKE THERMAL EFFICIENCY=         .30618E+00
                       B S F C, g/kW,h =         .26651E+03
          FUEL CONSUMPTION RATE, kg/h =         .14994E+02
        EXHAUST STACK TEMPERATURE, K =         .77279E+03

        BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=         .63655E+01
         ADDITIONAL OUTPUT - STIRLING, kW=         .95483E+01
  RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =         .34082E+00
  STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=         .35815E+00
             FUEL SAVINGS - RANKINE,kg/h =         .15240E+01
            FUEL SAVINGS - STIRLING,kg/h =         .21755E+01
```

NOTCH 3

```
                         OPERATING CONDITIONS

                                  R P M =      570.00
                      COMPRESSION RATIO =       11.50
             INJECTION ADVANCE,DEGREES =       20.00
                  INLET PRESSURE (BARS) =        1.84
               INLET TEMPERATURE (K) =      401.50
               IGNITION DELAY,DEGREES =        7.00
                 HEAT TRANSFER OPTION =      1

                          MODEL PREDICTIONS

                    INDICATED WORK (J) =         .98795E+04
         HEAT LOST BY CONVECTION (J) =         .62021E+04
                    GAS RADIATION (J) =         .22689E+03
                   SOOT RADIATION (J) =         .93723E+03
          HEAT LOST BY RADIATION (J) =         .11641E+04
              IND THERMAL EFFICIENCY =         .35545E+00
         IND OUTPUT PER CYLINDER (KW) =         .46928E+02
                       I M E P (PSI)=         .13090E+03
                       I M E P (BAR)=         .90254E+01
                       F M E P (PSI)=         .24749E+02
                       B M E P (PSI)=         .10616E+03
        BRAKE OUTPUT PER CYLINDER,kW =         .38055E+02
       BRAKE OUTPUT PER CYLINDER,BHP =         .51012E+02
              BRAKE THERMAL EFFICIENCY=         .28825E+00
                       B S F C, g/kW,h =         .28309E+03
          FUEL CONSUMPTION RATE, kg/h =         .10773E+02
        EXHAUST STACK TEMPERATURE, K =         .79501E+03

        BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=         .48408E+01
         ADDITIONAL OUTPUT - STIRLING, kW=         .72612E+01
  RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =         .32492E+00
  STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=         .34325E+00
             FUEL SAVINGS - RANKINE,kg/h =         .12157E+01
            FUEL SAVINGS - STIRLING,kg/h =         .17262E+01
```

NOTCH 2

```
                                   OPERATING CONDITIONS

                                   R P M =      472.00
                        COMPRESSION RATIO =      11.50
                INJECTION ADVANCE,DEGREES =      20.00
                    INLET PRESSURE (BARS) =       1.52
                    INLET TEMPERATURE (K) =     401.50
                   IGNITION DELAY,DEGREES =       7.00
                     HEAT TRANSFER OPTION =     1

                                   MODEL PREDICTIONS

                      INDICATED WORK (J) =          .67761E+04
           HEAT LOST BY CONVECTION (J) =            .48024E+04
                      GAS RADIATION (J) =           .17795E+03
                     SOOT RADIATION (J) =           .81437E+03
           HEAT LOST BY RADIATION (J) =             .99232E+03
                 IND THERMAL EFFICIENCY =           .35147E+00
          IND OUTPUT PER CYLINDER (KW) =            .26653E+02
                          I M E P (PSI)=            .89785E+02
                          I M E P (BAR)=            .61903E+01
                          F M E P (PSI)=            .24930E+02
                          B M E P (PSI)=            .64855E+02
          BRAKE OUTPUT PER CYLINDER,kW =            .19252E+02
         BRAKE OUTPUT PER CYLINDER,BHP =            .25807E+02
               BRAKE THERMAL EFFICIENCY=            .25388E+00
                         B S F C, g/kW,h =          .32141E+03
          FUEL CONSUMPTION RATE, kg/h =             .61879E+01
         EXHAUST STACK TEMPERATURE, K =             .72667E+03

         BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=          .24610E+01
         ADDITIONAL OUTPUT - STIRLING, kW=          .36915E+01
   RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =        .28633E+00
   STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=        .30256E+00
            FUEL SAVINGS - RANKINE,kg/h =           .70135E+00
           FUEL SAVINGS - STIRLING,kg/h =           .99560E+00
```

NOTCH 1

```
                                   OPERATING CONDITIONS

                                   R P M =      362.00
                        COMPRESSION RATIO =      11.50
                INJECTION ADVANCE,DEGREES =      20.00
                    INLET PRESSURE (BARS) =       1.16
                    INLET TEMPERATURE (K) =     401.50
                   IGNITION DELAY,DEGREES =       7.00
                     HEAT TRANSFER OPTION =     1

                                   MODEL PREDICTIONS

                      INDICATED WORK (J) =          .47697E+04
           HEAT LOST BY CONVECTION (J) =            .36568E+04
                      GAS RADIATION (J) =           .18934E+03
                     SOOT RADIATION (J) =           .90518E+03
           HEAT LOST BY RADIATION (J) =             .10945E+04
                 IND THERMAL EFFICIENCY =           .33785E+00
          IND OUTPUT PER CYLINDER (KW) =            .16388E+02
                          I M E P (PSI)=            .63199E+02
                          I M E P (BAR)=            .43573E+01
                          F M E P (PSI)=            .26054E+02
                          B M E P (PSI)=            .37145E+02
          BRAKE OUTPUT PER CYLINDER,kW =            .84568E+01
         BRAKE OUTPUT PER CYLINDER,BHP =            .11336E+02
               BRAKE THERMAL EFFICIENCY=            .19857E+00
                         B S F C, g/kW,h =          .41094E+03
          FUEL CONSUMPTION RATE, kg/h =             .34752E+01
         EXHAUST STACK TEMPERATURE, K =             .72529E+03

         BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=          .14160E+01
         ADDITIONAL OUTPUT - STIRLING, kW=          .21240E+01
   RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =        .23182E+00
   STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=        .24844E+00
            FUEL SAVINGS - RANKINE,kg/h =           .49843E+00
           FUEL SAVINGS - STIRLING,kg/h =           .69762E+00
```

IDLE

```
                        OPERATING CONDITIONS

                                R P M =     362.00
                    COMPRESSION RATIO =      11.50
             INJECTION ADVANCE,DEGREES =     20.00
               INLET PRESSURE (BARS) =       1.16
               INLET TEMPERATURE (K) =     401.50
              IGNITION DELAY,DEGREES =       7.00
                 HEAT TRANSFER OPTION =    1

                        MODEL PREDICTIONS

                  INDICATED WORK (J) =         .19334E+04
          HEAT LOST BY CONVECTION (J) =         .24829E+04
                  GAS RADIATION (J) =         .59323E+02
                 SOOT RADIATION (J) =         .27335E+03
           HEAT LOST BY RADIATION (J) =         .33267E+03
              IND THERMAL EFFICIENCY =         .29215E+00
         IND OUTPUT PER CYLINDER (KW) =         .58323E+01
                       I M E P (PSI)=         .25618E+02
                       I M E P (BAR)=         .17662E+01
                       F M E P (PSI)=         .24363E+02
                       B M E P (PSI)=         .12549E+01
         BRAKE OUTPUT PER CYLINDER,kW =         .28570E+00
        BRAKE OUTPUT PER CYLINDER,BHP =         .38298E+00
              BRAKE THERMAL EFFICIENCY=         .14311E-01
                       B S F C, g/kW.h =         .57017E+04
            FUEL CONSUMPTION RATE, kg/h =         .16290E+01
         EXHAUST STACK TEMPERATURE, K =         .53491E+03

            BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=         .38025E+00
         ADDITIONAL OUTPUT - STIRLING, kW=         .57037E+00
   RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =         .33359E-01
   STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=         .42882E-01
             FUEL SAVINGS - RANKINE,kg/h =         .93013E+00
            FUEL SAVINGS - STIRLING,kg/h =         .10853E+01
```

ANNEX B

PERFORMANCE OF THE LOW HEAT ENGINE WITH AND WITHOUT BOTTOMING CYCLES AT DIFFERENT THROTTLE NOTCH POSITIONS

NOTCH 8

```
                                  OPERATING CONDITIONS

                                  R P M =     1097.00
                      COMPRESSION RATIO =       11.50
             INJECTION ADVANCE,DEGREES =       20.00
                 INLET PRESSURE (BARS) =        3.53
                 INLET TEMPERATURE (K) =      401.50
              IGNITION DELAY,DEGREES =          7.00
                  HEAT TRANSFER OPTION =     1

                                  MODEL PREDICTIONS

                   INDICATED WORK (J) =          .21237E+05
          HEAT LOST BY CONVECTION (J) =          .52789E+04
                   GAS RADIATION (J) =           .16617E+03
                  SOOT RADIATION (J) =           .64295E+03
          HEAT LOST BY RADIATION (J) =           .80912E+03
             IND THERMAL EFFICIENCY =            .39684E+00
        IND OUTPUT PER CYLINDER (KW) =           .19414E+03
                       I M E P (PSI)=            .28139E+03
                       I M E P (BAR)=            .19401E+02
                       F M E P (PSI)=            .20498E+02
                       B M E P (PSI)=            .26089E+03
        BRAKE OUTPUT PER CYLINDER,kW =           .18000E+03
       BRAKE OUTPUT PER CYLINDER,BHP =           .24128E+03
            BRAKE THERMAL EFFICIENCY=            .36793E+00
                     B S F C, g/kW.h =           .22178E+03
         FUEL CONSUMPTION RATE, kg/h =           .39920E+02
       EXHAUST STACK TEMPERATURE, K =            .82922E+03

           BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=        .20533E+02
        ADDITIONAL OUTPUT - STIRLING, kW=        .30799E+02
  RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =      .40990E+00
  STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=      .43089E+00
              FUEL SAVINGS - RANKINE,kg/h =      .40874E+01
            FUEL SAVINGS - STIRLING,kg/h =       .58326E+01
```

NOTCH 7

```
                                  OPERATING CONDITIONS

                                  R P M =      987.00
                      COMPRESSION RATIO =       11.50
             INJECTION ADVANCE,DEGREES =       20.00
                 INLET PRESSURE (BARS) =        3.18
                 INLET TEMPERATURE (K) =      401.50
              IGNITION DELAY,DEGREES =          7.00
                  HEAT TRANSFER OPTION =     1

                                  MODEL PREDICTIONS

                   INDICATED WORK (J) =          .19774E+05
          HEAT LOST BY CONVECTION (J) =          .50455E+04
                   GAS RADIATION (J) =           .19683E+03
                  SOOT RADIATION (J) =           .75348E+03
          HEAT LOST BY RADIATION (J) =           .95031E+03
             IND THERMAL EFFICIENCY =            .39213E+00
        IND OUTPUT PER CYLINDER (KW) =           .16264E+03
                       I M E P (PSI)=            .26200E+03
                       I M E P (BAR)=            .18064E+02
                       F M E P (PSI)=            .21896E+02
                       B M E P (PSI)=            .24011E+03
        BRAKE OUTPUT PER CYLINDER,kW =           .14905E+03
       BRAKE OUTPUT PER CYLINDER,BHP =           .19979E+03
            BRAKE THERMAL EFFICIENCY=            .35936E+00
                     B S F C, g/kW.h =           .22707E+03
         FUEL CONSUMPTION RATE, kg/h =           .33844E+02
       EXHAUST STACK TEMPERATURE, K =            .86187E+03

           BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=        .18511E+02
        ADDITIONAL OUTPUT - STIRLING, kW=        .27767E+02
  RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =      .40399E+00
  STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=      .42630E+00
              FUEL SAVINGS - RANKINE,kg/h =      .37390E+01
            FUEL SAVINGS - STIRLING,kg/h =       .53150E+01
```

NOTCH 6

```
                              OPERATING CONDITIONS

                                 R P M =     889.00
                     COMPRESSION RATIO =      11.50
           INJECTION ADVANCE,DEGREES =      20.00
                INLET PRESSURE (BARS) =       2.86
               INLET TEMPERATURE (K) =     401.50
              IGNITION DELAY,DEGREES =       7.00
                HEAT TRANSFER OPTION =     1

                              MODEL PREDICTIONS

                 INDICATED WORK (J) =          .17349E+05
        HEAT LOST BY CONVECTION (J) =          .44781E+04
                 GAS RADIATION (J) =          .19382E+03
                SOOT RADIATION (J) =          .78339E+03
         HEAT LOST BY RADIATION (J) =          .97721E+03
             IND THERMAL EFFICIENCY =          .39324E+00
       IND OUTPUT PER CYLINDER (KW) =          .12853E+03
                      I M E P (PSI)=          .22987E+03
                      I M E P (BAR)=          .15849E+02
                      F M E P (PSI)=          .22531E+02
                      B M E P (PSI)=          .20734E+03
       BRAKE OUTPUT PER CYLINDER,kW =          .11593E+03
     BRAKE OUTPUT PER CYLINDER,BHP =          .15540E+03
           BRAKE THERMAL EFFICIENCY=          .35469E+00
                   B S F C, g/kW,h =          .23006E+03
         FUEL CONSUMPTION RATE, kg/h =          .26670E+02
      EXHAUST STACK TEMPERATURE, K =          .85619E+03

          BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

         ADDITIONAL OUTPUT - RANKINE, kW=          .14663E+02
        ADDITIONAL OUTPUT - STIRLING, kW=          .21994E+02
RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =          .39956E+00
STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=          .42199E+00
           FUEL SAVINGS - RANKINE,kg/h =          .29945E+01
          FUEL SAVINGS - STIRLING,kg/h =          .42530E+01
```

NOTCH 5

```
                              OPERATING CONDITIONS

                                 R P M =     779.00
                     COMPRESSION RATIO =      11.50
           INJECTION ADVANCE,DEGREES =      20.00
                INLET PRESSURE (BARS) =       2.50
               INLET TEMPERATURE (K) =     401.50
              IGNITION DELAY,DEGREES =       7.00
                HEAT TRANSFER OPTION =     1

                              MODEL PREDICTIONS

                 INDICATED WORK (J) =          .15581E+05
        HEAT LOST BY CONVECTION (J) =          .41444E+04
                 GAS RADIATION (J) =          .22789E+03
                SOOT RADIATION (J) =          .92720E+03
         HEAT LOST BY RADIATION (J) =          .11551E+04
             IND THERMAL EFFICIENCY =          .38811E+00
       IND OUTPUT PER CYLINDER (KW) =          .10115E+03
                      I M E P (PSI)=          .20646E+03
                      I M E P (BAR)=          .14234E+02
                      F M E P (PSI)=          .23798E+02
                      B M E P (PSI)=          .18266E+03
       BRAKE OUTPUT PER CYLINDER,kW =          .89490E+02
     BRAKE OUTPUT PER CYLINDER,BHP =          .11996E+03
           BRAKE THERMAL EFFICIENCY=          .34337E+00
                   B S F C, g/kW,h =          .23764E+03
         FUEL CONSUMPTION RATE, kg/h =          .21267E+02
      EXHAUST STACK TEMPERATURE, K =          .88825E+03

          BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

         ADDITIONAL OUTPUT - RANKINE, kW=          .12387E+02
        ADDITIONAL OUTPUT - STIRLING, kW=          .18581E+02
RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =          .39090E+00
STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=          .41467E+00
           FUEL SAVINGS - RANKINE,kg/h =          .25858E+01
          FUEL SAVINGS - STIRLING,kg/h =          .36564E+01
```

NOTCH 4

```
                              OPERATING CONDITIONS

                                   R P M =     680.00
                       COMPRESSION RATIO =      11.50
              INJECTION ADVANCE,DEGREES =      20.00
                   INLET PRESSURE (BARS) =       2.20
                   INLET TEMPERATURE (K) =     401.50
                IGNITION DELAY,DEGREES =        7.00
                   HEAT TRANSFER OPTION =     1

                              MODEL PREDICTIONS

                     INDICATED WORK (J) =         .12716E+05
            HEAT LOST BY CONVECTION (J) =         .34259E+04
                     GAS RADIATION (J) =          .19843E+03
                    SOOT RADIATION (J) =          .90246E+03
             HEAT LOST BY RADIATION (J) =         .11009E+04
                 IND THERMAL EFFICIENCY =         .39216E+00
           IND OUTPUT PER CYLINDER (KW) =         .72059E+02
                           I M E P (PSI)=         .16849E+03
                           I M E P (BAR)=         .11617E+02
                           F M E P (PSI)=         .23975E+02
                           B M E P (PSI)=         .14452E+03
           BRAKE OUTPUT PER CYLINDER,kW =         .61806E+02
          BRAKE OUTPUT PER CYLINDER,BHP =         .82849E+02
               BRAKE THERMAL EFFICIENCY=          .33636E+00
                        B S F C, g/kW,h =         .24260E+03
            FUEL CONSUMPTION RATE, kg/h =         .14994E+02
           EXHAUST STACK TEMPERATURE, K =         .85486E+03

              BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

             ADDITIONAL OUTPUT - RANKINE, kW=      .84939E+01
            ADDITIONAL OUTPUT - STIRLING, kW=      .12741E+02
   RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =      .38258E+00
   STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=      .40569E+00
              FUEL SAVINGS - RANKINE,kg/h =       .18116E+01
             FUEL SAVINGS - STIRLING,kg/h =       .25627E+01
```

NOTCH 3

```
                              OPERATING CONDITIONS

                                   R P M =     570.00
                       COMPRESSION RATIO =      11.50
              INJECTION ADVANCE,DEGREES =      20.00
                   INLET PRESSURE (BARS) =       1.84
                   INLET TEMPERATURE (K) =     401.50
                IGNITION DELAY,DEGREES =        7.00
                   HEAT TRANSFER OPTION =     1
                              MODEL PREDICTIONS

                     INDICATED WORK (J) =         .10723E+05
            HEAT LOST BY CONVECTION (J) =         .30445E+04
                     GAS RADIATION (J) =          .22772E+03
                    SOOT RADIATION (J) =          .10703E+04
             HEAT LOST BY RADIATION (J) =         .12980E+04
                 IND THERMAL EFFICIENCY =         .38579E+00
           IND OUTPUT PER CYLINDER (KW) =         .50933E+02
                           I M E P (PSI)=         .14208E+03
                           I M E P (BAR)=         .97958E+01
                           F M E P (PSI)=         .25252E+02
                           B M E P (PSI)=         .11683E+03
           BRAKE OUTPUT PER CYLINDER,kW =         .41881E+02
          BRAKE OUTPUT PER CYLINDER,BHP =         .56140E+02
               BRAKE THERMAL EFFICIENCY=          .31722E+00
                        B S F C, g/kW,h =         .25723E+03
            FUEL CONSUMPTION RATE, kg/h =         .10773E+02
           EXHAUST STACK TEMPERATURE, K =         .87839E+03

              BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

             ADDITIONAL OUTPUT - RANKINE, kW=      .63856E+01
            ADDITIONAL OUTPUT - STIRLING, kW=      .95784E+01
   RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =      .36559E+00
   STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=      .38978E+00
              FUEL SAVINGS - RANKINE,kg/h =       .14253E+01
             FUEL SAVINGS - STIRLING,kg/h =       .20053E+01
```

NOTCH 2

NOT AVAILABLE

NOTCH 1

```
                                     OPERATING CONDITIONS

                                            R P M =      362.00
                                COMPRESSION RATIO =       11.50
                       INJECTION ADVANCE,DEGREES  =       20.00
                          INLET PRESSURE (BARS)   =        1.16
                          INLET TEMPERATURE (K)   =      401.50
                        IGNITION DELAY,DEGREES    =        7.00
                           HEAT TRANSFER OPTION   =      1

                                     MODEL PREDICTIONS

                            INDICATED WORK (J) =        .53085E+04
               HEAT LOST BY CONVECTION (J) =            .16746E+04
                            GAS RADIATION (J) =         .16751E+03
                           SOOT RADIATION (J) =         .10408E+04
                 HEAT LOST BY RADIATION (J) =           .12083E+04
                      IND THERMAL EFFICIENCY =          .37602E+00
              IND OUTPUT PER CYLINDER (KW) =            .16014E+02
                                  I M E P (PSI)=        .70339E+02
                                  I M E P (BAR)=        .48496E+01
                                  F M E P (PSI)=        .26375E+02
                                  B M E P (PSI)=        .43964E+02
             BRAKE OUTPUT PER CYLINDER,kW =             .10009E+02
            BRAKE OUTPUT PER CYLINDER,BHP =             .13417E+02
                    BRAKE THERMAL EFFICIENCY=           .23502E+00
                              B S F C, g/kW.h =         .34720E+03
              FUEL CONSUMPTION RATE, kg/h =             .34752E+01
             EXHAUST STACK TEMPERATURE, K =             .81617E+03

                     BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

               ADDITIONAL OUTPUT - RANKINE, kW=         .20263E+01
              ADDITIONAL OUTPUT - STIRLING, kW=         .30395E+01
     RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =          .28260E+00
     STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=          .30639E+00
                FUEL SAVINGS - RANKINE,kg/h =           .58509E+00
                FUEL SAVINGS - STIRLING,kg/h =          .80949E+00
```

IDLE

```
                                  OPERATING CONDITIONS

                                   R P M =      362.00
                        COMPRESSION RATIO =      11.50
                INJECTION ADVANCE,DEGREES =      20.00
                    INLET PRESSURE (BARS) =       1.16
                  INLET TEMPERATURE (K) =       401.50
                IGNITION DELAY,DEGREES =          7.00
                    HEAT TRANSFER OPTION =     1

                                  MODEL PREDICTIONS

                       INDICATED WORK (J) =           .25557E+04
             HEAT LOST BY CONVECTION (J) =            .59306E+03
                       GAS RADIATION (J) =            .25226E+02
                      SOOT RADIATION (J) =            .32117E+03
              HEAT LOST BY RADIATION (J) =            .34639E+03
                  IND THERMAL EFFICIENCY =            .38620E+00
            IND OUTPUT PER CYLINDER (KW) =            .77098E+01
                           I M E P (PSI)=             .33864E+02
                           I M E P (BAR)=             .23348E+01
                           F M E P (PSI)=             .24734E+02
                           B M E P (PSI)=             .91304E+01
           BRAKE OUTPUT PER CYLINDER,kW =             .20787E+01
          BRAKE OUTPUT PER CYLINDER,BHP =             .27865E+01
                BRAKE THERMAL EFFICIENCY=             .10413E+00
                        B S F C, g/kW,h =             .78366E+03
           FUEL CONSUMPTION RATE, kg/h =              .16290E+01
         EXHAUST STACK TEMPERATURE, K =               .63184E+03

            BOTTOMING CYCLE SYSTEMS & THEIR PERFORMANCE

          ADDITIONAL OUTPUT - RANKINE, kW=            .83110E+00
         ADDITIONAL OUTPUT - STIRLING, kW=            .12467E+01
 RANKINE CYCLE - TOTAL SYSTEM EFFICIENCY =            .14576E+00
 STIRLING CYCLE - TOTAL SYSTEM EFFICIENCY=            .16657E+00
            FUEL SAVINGS - RANKINE,kg/h =             .46528E+00
           FUEL SAVINGS - STIRLING,kg/h =             .61070E+00
```

APPENDIX D

TRACTION MOTOR DEVELOPMENT ACTIVITIES

D.1 INTRODUCTION

This appendix is summarized from Research Report EE/EPG/86, prepared by Mr. Yaozhong Xu, under the supervision of Drs. V.I. John and G.E. Dawson, Department of Electrical Engineering, Queen's University at Kingston.

D.2 EXISTING TRACTION TECHNOLOGY

D.2.1 DC Traction Motors

DC series motors were the early choice for railway applications. Several reasons why dc series motors have been so greatly favoured for traction motive power applications are summarized below.

- The series field winding consists of a comparatively small number of turns of fairly large section copper with very low voltage between turns. The proportion of insulating material to copper can be kept low. Consequently, the winding is small and compact and has a high rate of heat transfer. Sufficiently rigid construction provides a great resistance to shock and vibration.

- Its main pole flux varies with (but not necessarily in proportion to) the armature current. The variation in torque will always be greater than the corresponding variations in current. [Series-wound, single-phase ac commutator motors also have similar torque characteristics. However, the problems associated with design, commutation, power factor and efficiency put this type of motor at a considerable disadvantage over its dc counterpart as a means of motive power for electric traction.]

- The torque corresponding to a given load current is entirely independent of variations in line voltage. As a result, for parallel operation of dc series motors on one locomotive or on a multiple-unit train, complete reliance can be placed on the current taken by each individual motor at any given train speed varying from that of any other motor within very close limits.

- In dc series motors, the speed automatically decreases as torque increases and this prevents overloading of the motor when operating on its full free-running characteristics. This self-protecting property is of immense value, specially in suburban railway service where motors can be subjected to wide voltage fluctuations and frequent interruptions.

- The control circuit and power-conditioning unit circuit for dc series motors are much simpler than those for ac motors. Either controlled rectifiers or dc choppers can be used, depending on whether the supply voltage is ac or dc.

Some of the main disadvantages of dc series motors are high initial cost, high maintenance cost due to the presence of commutator and brushes, and related problems (for example, they are less compact and have a greater probability of breakdown).

Other types of dc motors, for example shunt-type dc motors and separately excited dc motors, can also be used for traction vehicle applications. With a proper power electronic control circuit, they can be controlled to have characteristics similar to dc series motors. However, due to the presence of mechanical commutators and brushes, they all suffer from the same shortcomings: high cost of initial investment and maintenance, lower reliability, and most of all limited application for high-speed and high-power traction vehicles.

Commutation Problems of DC Traction

Commutation is not a new problem. As a matter of fact, it had been studied since the invention of the dc motor. Research has been conducted in this area and improvements in commutation performance observed in the following:

- the development of high-performance insulation materials
- the universal adoption of carbon graphite brushes
- the development of more advanced commutator and brush structures, with modifications in brush arrangement (for example, split brushes).

The present technological status is such that commutation becomes the bottleneck for high-speed, high-power dc traction motor applications. Following is a condensed summary of factors affecting commutation performance.

1. First is the effectiveness and type of wheel-slip control system, and related wheel-slip protection schemes. The sudden stop of a runaway axle can cause commutator flashover in a traction motor unless the power is cut off before the application of sand. A large wheel slip resulting from improper control will run away at much higher speed than is allowed by maximum designed values, which will also cause severe flashover of the commutator.

2. A very serious problem, which seems more prevalent with electric locomotives than with diesel engine locomotives, is the production of "sanding" or "stall" burns on commutators, occurring in traction motors which are subjected to heavy starting current with wheels locked for an excessive period. This phenomenon is amplified due to sufficient movement of the armature in the gear teeth clearance for the brushes to take up their new positions with the brushbox clearance so that they will now be in contact with the commutator along the lending edge only, presenting what is virtually a line contact with one individual commutator segment. The larger contact resistance assisted by heavy starting current greatly increases the chance of severe flashover or still burns on the otherwise smooth commutators and the insulation materials between the commutator segments. As

a result, the damaged commutator further deteriorates the commutation performance. This positive feedback effect can very quickly wear out new commutators and brushes.

3. Insufficient pressure between the commutator as well as mechanical vibration and shocks between rail track and locomotive wheels can cause frequent brush chattering, which will result in the following damage:

   (a) chattering of the brushes with consequent flashover damage

   (b) very bad sparking with resulting damage to the commutator surface and consequent formation of commutator flats which further deteriorate commutation performance

   (c) the production of free and coagulated copper particles from the commutator segments which can ultimately cause severe flashover damage.

4. The maximum speed limit is not always observed when the locomotive is manually operated, due to operator's negligence, or some other reason. Exceeding the designed maximum speed of a traction motor only once for a few minutes is sufficient to severely damage, and sometimes even permanently damage, the commutator, with such adverse results as short brush life, commutator flattening and flashovers.

Engineering practice has verified, in determining the ratings of dc traction motors, the importance of two design parameters: the average reactance voltage between the commutation segments, U_{AR} (steady-state effect), and the induced voltage in the coils by the main pole-fringing field flux in the commutation zone and shorted by the brushes (transient effect), U_k. As shall be discussed in the section on slotless dc traction motors, both U_{AR} and U_k include four main components:

(1) winding end flux
(2) armature core teeth and slot flux
(3) leakage flux due to winding end banding materials
(4) interpolar space leakage flux.

The key factor in improving the commutation performance is to reduce these leakage flux components to a minimum. Item (3) can be reduced to a minimum if non-ferromagnetic materials are used for bandings. Item (4) is absent if there is no interpolar space. Item (1) cannot be eliminated, but with proper coil end turn shaping its effect can reduced to a minimum. Item (2) is totally eliminated if slotless armature dc motors are used.

D.2.2 AC Traction Motors

Asynchronous Motors (Squirrel-Cage Induction Motors)

Recent advances in solid-state converter technology have made it possible to consider the application of the squirrel-case induction motor (SCIM) for traction purposes. It is predicted that the introduction of GTO thyristors in converter technology will soon make it possible to replace dc series motors in 50/60/16^{2}/3 Hz single-phase traction systems or in diesel-electric traction systems.

The main advantages of the converter-fed, three-phase, squirrel-cage induction motor traction drives can be summarized as follows:

1. The motors are lightweight, compact and rugged. Due to the lack of commutator and brushes, the induction motor weighs considerably less than the usual single-phase series-wound commutator motor or the dc motor (approximated in the proportion 1:2:1.5).[1] As a result, it is possible to install more power per wheelset, which is a very important fact for building high-power traction vehicles. Moreover, rail wear caused by high-speed vehicles will be reduced as a result of smaller unsprung masses in the bogies.

2. There is high reliability and considerably less maintenance because of the lack of commutator and brushes, considered vital in traction motive power applications.

3. Maximum torque can be maintained at any speed, including starting and standstill.

4. The torque versus speed characteristics of the induction motor in the field-weakening range declines inversely proportional to speed, while the torque of the single-phase series-wound commutator motor decreases inversely proportional to the square of the speed. Hence, the induction motor drive speed system can provide more tractive force at high speed. This property is very important, since it makes it possible to haul high-speed passenger trains as well as heavy freight trains without changing the gear-box ratio.[2]

5. Motor windings can be designed for higher voltages, resulting in a more favourable design of other components such as transformer secondary, circuit breakers, wiring, etc., in high-power vehicles.

6. Because of the steep torque versus speed characteristics of an induction motor fed with a certain frequency, there will be no spinning of one or more single wheelsets. Hence it is possible to fully utilize the increased power installation per wheelset. The BBC-built DE2500 type locomotive has demonstrated this in a very convincing manner by starting and hauling freight trains of 2500t up to 3500t load.[3,4,5]

Due to the high cost and bulkiness of the mechanical and rotary frequency conversion units, earlier efforts, before the advent of thyristors, to apply induction motors for traction purposes were not successful.

The wound-rotor (slip-ring) induction motor can also be used for traction vehicle applications. However, compared with the squirrel-case induction motor, it is less compact due to the presence of slip-rings and rotor windings. It does offer some added advantages over the squirrel-cage induction motor in that it can be controlled from the rotor also.

Synchronous Motors

One of the main disadvantages of the dc motor is the presence of mechanical commutators and brushes which cause commutation problems at heavy loads and incur high maintenance cost. One method to eliminate this problem is through the use of an electronic commutator which leads to the concept of brushless dc drives using a synchronous motor. One of the proven configurations of the brushless system consists of a two-quadrant, transistorized front-end chopper and a three-phase transistorized inverter, which form the electronic commutator for the synchronous machine phase current. Some research work[6,7,8,9,10,11,12,13,14,15] has been carried out in this area. With proper electronic control, the system can operate stably for all speeds and avoid loss of synchronization. Compared with dc drive systems, the low-speed performance of brushless dc drives can be further improved by proper switching of the electronic commutator.

It is interesting to compare an ordinary dc drive system with a brushless dc drive system. In both cases there exist two independent flux generation mechanisms. One is through the armature circuit and the other is through the field circuit. The dc machine is constructed with high power armature as the rotary member, with power connection through carbon brushes and the copper commutators, which transform the dc supply into a proper ac supply in order to produce a rotating field at the rotor speed. In a synchronous machine, the design allows the rotor to carry the low power field, thus the high power armature (stator) can be directly connected to a three-phase ac supply which can produce a synchronously rotating field without any mechanical commutation aids.

The two obvious advantages of synchronous motors over dc motors are that the synchronous motors require less maintenance, and they can be built for much higher power rating. Both factors are important for high-power traction vehicle applications.

Some other features of brushless motors are as follows:

- high efficiency due to thyristorized speed controller
- performance for four-quadrant running is competitive to dc machine drives controlled by solid-state, totally adjustable, reversing control and is superior in its simplicity of control.

Serious consideration has been given to synchronous motors in the area of a brushless drive system to replace the ordinary dc motor drive system, especially in France[16] and some other European countries,[17,18,19,20] as well as in the Soviet Union.[21,22]

D.3 ACTIVE TRACTION MOTOR RESEARCH

Two recent technological innovations, the slotless dc motor and the permanent magnet synchronous motor, have been considered for application as high-power, high-speed traction motors. These technologies are still in the research and development stage, but the preliminary results have been promising.

D.3.1 Slotless DC Motors

The major limiting factor of the power and torque rating for a dc traction motor of given size is commutation performance. There are a number of factors which affect commutation performance:

- the reactance emf between commutator segments
- the transformer emf
- load current
- motor peripheral speed
- contact pressure between brush and commutator segments
- mechanical instability caused by force inputs from the track to the locomotive.

Historically, many electrical machines constructed before 1910 used slotless (or surface winding) structures. It was recognized at that time that the use of slots made commutation more difficult. However, a number of mechanical and materials problems led to the universal adoption of slotted windings. The major advantages were:

- the coils could be securely fastened to resist radial and tangential forces
- the low thermal resistance of the core materials used for slotted windings helped to reduce winding temperatures
- the ferromagnetic materials then available to anchor the surface coil caused a severe deterioration of commutation performance due to increased leakage flux.

Recent technological developments have made a slotless structure practical and desirable:

- high-strength, non-ferromagnetic materials are available to anchor the surface windings of slotless dc motors
- modern insulation materials combining high mechanical strength and low thermal resistance have made slotted structure unnecessary from a thermal dissipation standpoint.

The average reactance emf developed between the commutator segments in the commutation zone is a good measurement of the severity of commutation conditions. This reactance emf is produced by the quasi-static magnetic field in the area. This quasi-static magnetic field contains the following components:

- end winding leakage flux (about 1/4 to 1/3)
- slot leakage flux (about 2/3 to 3/4)
- band leakage flux (only if magnetic banding wire were used)
- interpolar space leakage flux.

The elimination of slots completely eliminates the slot leakage flux, and the corresponding average reactance emf component due to slot leakage flux. The reduction in the total leakage flux in the commutation zone reduces the level of interpole flux required to totally eliminate the flux in the commutation zone, and alleviates interpole saturation problems. The uniform arrangement of surface windings reduces the difference in reactance emf between the commutator segments, so that better compensation can be provided by interpole windings.

However, while a slotless structure can greatly improve commutation performance, it does introduce a number of new problems, which are being addressed by ongoing research. The Soviet Union was an early leader in this area, notably with the results of Bocharov.[23,24,25] Great Britain started its research program somewhat later. In 1981, work on an experimental 90 hp slotless motor sponsored by British Rail (BR) was begun. The project was completed in 1984.[26,27,28,29] In 1982, BR's R&D Centre at Derby undertook the conversion of five dc traction motors of 340 hp continuous rating to a slotless structure. Results of laboratory testing and experimentation revealed a number of problems with the slotless motor, notably:

- inferior tractive effort at low motor speeds
- slightly (5 percent) higher operating temperatures
- very high levels of shear stress (up to 2 MN/m) between winding and core during short-circuit faults.

The latter problem is of sufficient magnitude to render this class of motor unfeasible at present. Significant fundamental research will be required to develop an insulating system capable of withstanding these shear forces at working temperatures.

D.3.2 Permanent Magnet Motors

Permanent magnet synchronous motors offer many advantages in terms of ruggedness and reduced maintenance requirements, since the permanent magnet field eliminates the need for slip-rings and brushes and a field supply. The permanent magnet, however, must be designed to withstand the large demagnetizing force of the stator currents. With recent improvements in magnet strength and quality, permanent magnet excitation has been successfully applied to motors of small and medium rating. Permanent magnet synchronous motors, like squirrel-cage induction motors, are rugged, small and require little maintenance. Consequently, this technology is potentially attractive for high-power, high-speed traction applications. The permanent magnet synchronous motor also possesses all the properties of the synchronous motor, including precise speed-matching capability between different motors connected in parallel or in series, and they can also be operated at a leading power factor. However, the feasibility of this approach for high-horsepower (1,000 hp plus) applications has not yet been established. The presence of field-induced voltage or back emf even under no-load conditions simplifies the converter structure used to supply the motor, since the natural line commutation can be used, instead of forced commutation as is necessary in converter-fed induction motor drives.

However, permanent magnet synchronous motors are much more expensive than conventional synchronous motors due to the high cost of permanent magnets. Also, because the field is provided by a constant-strength permanent magnet, field weakening is more difficult, and the range of feasible field weakening is quite limited.

Some research work has been carried out in the area of permanent magnet synchronous motor drives[30,31,32,33] but these investigations have been more concerned with the theoretical analysis of steady-state and dynamic performance of permanent magnet synchronous motor drives.

D.3.2.1 Constant Torque Operation

Neglecting the small stator resistance, R_s, the internally developed torque can be expressed as:

$$Te = 3 \left(\frac{P}{2}\right) I_s I_r L_m \sin \beta$$

For constant torque operation $I_s \sin \beta$ must be kept constant. The resulting power factor is also kept constant if I_s and β are kept constant individually, which corresponds to constant flux operation. The motor speed is, therefore, linearly proportional to the stator voltage applied. In this respect there is not much difference between the conventional synchronous motor drive and the permanent magnet synchronous motor drive.

D.3.2.2 Constant Power Operation

From the power equation it is clear that there are at least two ways to keep the motor output power constant:

- β is kept constant and (ωI_s) is kept constant
- I_s is kept constant and ($\omega \sin \beta$) is kept constant.

For the first method, the speed is inversely proportional to the stator current. Since $I_m = I_s + I_r$, to increase the motor rotational speed, I_s is reduced. The magnitude of I_m will increase or decrease, depending on the initial operating values of β. A decrease in I_m results in field weakening, an increase in I_m is just the opposite. As a result of the latter phenomenon, the stator terminal voltage, U_r, is increased. The speed variation using this method is also limited by the maximum stator voltage.

The second method works in a similar manner. To increase the motor opening speed, $\sin \beta$ must be reduced. If $\beta < 90°$, beta must be reduced, as a result I_m is increased. If $\beta > 90°$, beta must be increased, as a result I_m is decreased. For the reasons discussed above, this second method is preferred if β is initially operating at greater than 90° region, since in this case an increase in speed is accomplished by a decrease in the resultant airgap flux.

D.3.2.3 Pullout Torque

The maximum torque is limited by $I_s \sin \beta$, and for $\beta = \frac{\pi}{2}$ the pullout torque is proportional to I_s, which is principally limited by the armature conductor thermal capability.

It is interesting to note from the torque expression given above that the internal developed torque is independent of speed. It is a function of $\sin \beta$ and I_s only. For $\beta > 90°$ the operation is unstable in nature. However, with feedback control, stable operation in this region is possible. In this case, the motor is operating at either a leading or a lagging power factor and the resulting I_m is smaller than that in the operating region of $\beta < 90°$. The critical value of β at which the system is operating at unit power factor (that is $0 = 0°$) can be computed, assuming R_s is negligible, using the following expression:

$$\mathrm{ctan}\beta_0 + \frac{I_s}{I_r \sin \beta_0} = \tan \emptyset = 0$$

Assuming $\beta \neq 0°$ or 180°, the expression reduces to:

$$B = \cos^{-1}\left(-\frac{I_s}{I_r}\right)$$

Clearly, unity power factor operation is possible only when:

$$|I_s| \leq |I_r|$$

Leading power factor operation is ensured when $\beta \geq \beta_0$.

If R_s is not neglected, then the expression for the critical β is more complicated, but it can be derived from the power factor equation:

$$\frac{\omega L}{R_s \sin \beta} \left(\frac{L_s}{I_r} + \cos \beta \right) = 0$$

Again assuming $\beta = 0°$ or $180°$, β is given by:

$$\sin (\beta - \phi) = \frac{\omega L \quad I_s}{\sqrt{R_s^2 + (\omega L)^2} \; I_r}$$

where $\phi = \tan^{-1} \left(\frac{\omega L}{R_s} \right)$

It is much more complicated to find the leading power factor operating region than in the previous case, in which R_s is neglected. Also, it is interesting to note that unity power factor operation is not always possible for permanent magnet synchronous motor drive systems fed from a current source inverter.

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APPENDIX E

DESCRIPTION OF THE DYNAMIC CURVING MODEL "RACES"

This program is a general-purpose dynamic curving simulation which predicts the performance of a rail vehicle starting from tangent track and traversing an ideal clothoid spiral followed by a circular curve. RACES includes non-linear wheel-rail geometry with one or two points of contact and two methods of calculating creep forces. It also features automatic generation of the equations-of-motion, providing for non-linear suspension elements, including a dry-friction pad and a bi-linear spring.

The program consists of several modules which can be used to construct mathematical models for a variety of rail vehicle configurations, including locomotives equipped with conventional, cross-braced and forced-steering trucks. The main modules of the program are:

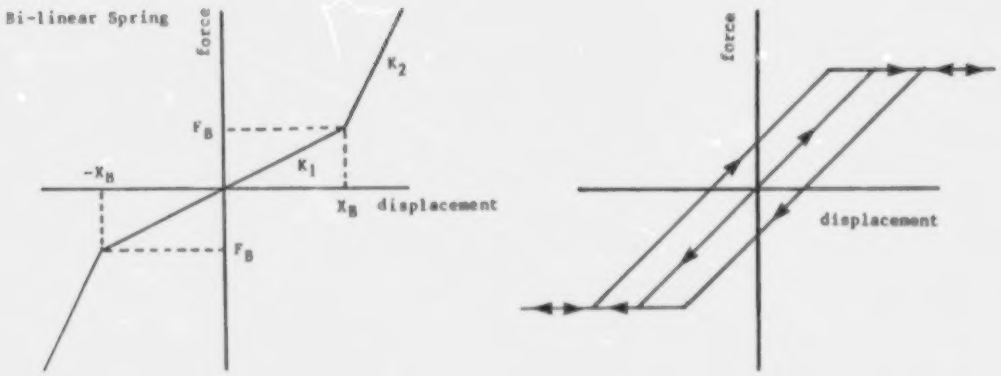
- a vehicle equation-of-motion generator
- a load transfer calculator
- a wheel/rail geometry and forces preprocessor
- the curving simulator.

E.1 VEHICLE EQUATION-OF-MOTION GENERATOR

The vehicle equation-of-motion generator A'GEM[1,2] requires only a simple numerical input file to describe the vehicle. The numerical input file is built interactively by the user, based on the physical model representing the actual vehicle. The vehicle models are built up from rigid bodies interconnected by suspension elements. Only the equations-of-motion corresponding to the degrees-of-freedom required for the simulation are generated, the others being simply constrained at the discretion of the user. The suspension elements are parallel combinations of linear and non-linear springs and dampers. Each linear element is characterized by its stiffness and damping coefficient, the positions of its ends and the rigid bodies to which the ends are attached. For the non-linear elements, other characteristics such as breakaway force and friction coefficient are also required. The types of suspension elements that can be modelled are:

- linear springs and dampers
- bi-linear springs
- frictional sliders
- torsional springs and dampers.

Examples of the bi-linear spring and the frictional slider models are shown in Figure E.1. These elements are useful for modelling suspensions which include dead-band elements, bump stops, and other similar components.



(a) (b)

Figure E.1 Representation of non-linear suspension elements

E.2 LOAD TRANSFER CALCULATOR

The transfer of load from one wheelset to another when axle torques are applied has a significant effect on the distribution of limiting traction characteristics among the wheelsets. The reaction to a torque applied to an axle by a motor suspended from the vehicle must be an equal and opposite torque applied to the vehicle above the primary suspension. The torques applied to the vehicle, through the motor mounts, affect the manner in which the total weight of the vehicle is shared by the axles. This results in an unequal distribution of limiting traction forces among the axles.

The load transfer is calculated from a linear structural model of the vehicle which accounts for all the significant vertical and longitudinal suspension elements. The governing equations are automatically derived using the A'GEM program described above. Individual torques are specified for each of the axles on the vehicle. From a knowledge of the locations of the motors and the reductions of the gearboxes, it is a relatively simple matter to determine the torques provided by the motors and the resulting forces which are transmitted through their supports. These forces are then used as input to the structural model, and the change in vertical reaction force seen at each axle is calculated. Since this is a linear calculation, a set of scaling factors can be readily established to determine the effect of other sets of axle torques.

E.3 WHEEL/RAIL GEOMETRY AND FORCES

A number of geometric parameters are required to adequately simulate the wheel-rail geometry and the resulting wheel-rail forces. It is assumed in the model that the wheel and the rail are always in contact and therefore no wheel lift is possible. Consequently, the wheelset motion is constrained to follow the wheel and rail geometries. Due to the non-linear wheel-rail profiles commonly used, the wheelset roll angle, the wheel-rail contact angle and the rolling radius all depend on the position of the wheelset on the rails. Only two variables are required to completely define the constrained wheelset position on the rails: the lateral displacement and the yaw angle. The wheelset yaw angle has only a second-order effect on the position of the points of contact and is therefore neglected. It is thus assumed that all contact parameters are functions of the wheelset lateral displacements only.

The wheel-rail geometry is complex and requires accurate modelling. It is therefore impossible to derive analytical expressions for any cases except the simplest, and numerical analysis is required. The computer program WHRAILA[3] is used to characterize non-linear wheel-rail profiles. To run with RACES, the output of WHRAILA must be modified by a wheel-rail geometry preprocessor which generates a wheel-rail geometry data file as a function of the wheelset lateral displacement.

To predict the wheel-rail contact forces, two methods are used: a cubic saturation model, and the FASTSIM algorithm published by Kalker.[4] In general, the cubic saturation model is a simple approximation of a very complex problem. It is very efficient, but should be used with caution. On the other hand, the slower FASTSIM can be used with confidence to calculate creep forces, since its results have been shown to agree well with available experimental data. The simulation results obtained with the cubic saturation model must be compared with those obtained with FASTSIM to prove their validity. In practice to date, the cubic saturation model has proven to be reliable and requires approximately 30 percent less computing effort than FASTSIM.

E.4 CURVING SIMULATOR

To predict accurately the rail vehicle motion while travelling from tangent track, through the transition and into the constant radius curve, the track geometry must be defined carefully. In the clothoid transition, the track curvature varies linearly from zero on the tangent track to the curvature of the circular curve which is equal to the inverse of its radius. The superelevation also varies linearly along the transition, starting from zero at the transition entry to reach a maximum at the entry of the circular curve, after which it is held constant.

The number of axles can vary from two to six, and axle torques are assumed constant. The vehicle is started on tangent track. During a run, the user has the option of modifying the load transfer/applied torque data, the tread and flange friction coefficients, and the method of creep force calculation. A variable-step Runge-Kutta integrator is used to obtain the time solution of the equations-of-motion.

Output from the RACES simulation is presented both in numerical and plotted forms. A number of plots can be drawn to analyze the data. All plots are presented as functions of distance of the leading truck centre along the track alignment. For a simulation, the following data can be plotted by the companion plotting program:

(1) lateral displacement
(2) angle of attack
(3) interaxle angle and alignment index
(4) flange forces
(5) lateral to vertical wheel-rail forces ratio
(6) saturation index for each wheel
(7) frictional work per axle
(8) non-linear element forces
(9) selected internal displacements.

The user can modify the program to have any of the variables printed as a function of time. Given the large number of internal forces, accelerations, etc. which are available at each instant of simulated time, the simulated results can be put in any form required.

E.5 VEHICLE MODELS

The vehicle models are generated in text files from which the relevant equations of motion are generated. This allows great flexibility in model configuration. For the load transfer and curving models, the rigid bodies are assumed to be:

- the carbody
- the truck frame
- the wheelsets.

For the curving model, the traction motor masses and inertias are lumped with the wheelsets. For the purpose of load transfer calculations, the traction motor reaction forces are assumed to act directly on the truck frame. Simulation tests have shown that this results in the same load transfer values as when the traction motors are modelled as separate bodies.

The six-axle locomotive is modelled as a nine-body, 27 degree-of-freedom system (Figure E.2). The degrees of freedom of each free body are shown in Table E.1. The spin degree of freedom assigned to the wheelsets allows the superposition of axle torques, an important parameter for accurate locomotive simulation. A similar configuration was used for the four-axle locomotives.

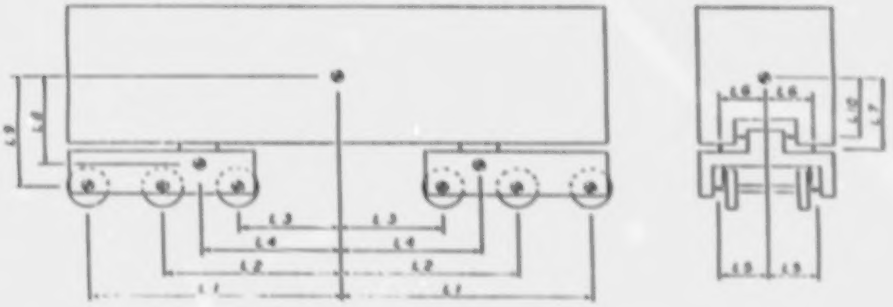


Figure E.2 Six-axle locomotive model

TABLE E.1

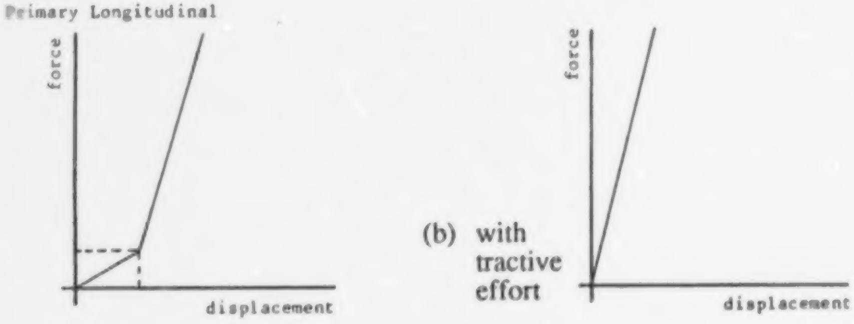
MODEL DEGREES-OF-FREEDOM

	Longitudinal Displacement	Lateral Displacement	Verical Displacement	Roll Angle	Pitch/Spin Angle	Yaw Angle
Curving model:						
wheelsets		x			x	
trucks		x		x		x
carbody		x		x		x
Load transfer model:						
wheelsets	x		x			
trucks	x		x	x		
carbody	x		x	x		

The mass and suspension values used were taken from Garg[5] and from private correspondence with EMD and Bombardier.

The assumptions made in modelling the vehicle suspension are:

(a) **Primary suspension**. The primary suspension points are assumed to be located at the ends of the axles. The longitudinal suspension is modelled as a bi-linear component, except when tractive effort is applied. Then it is assumed that the tractive forces move the axles against their longitudinal stops, which are modelled as stiff linear springs. Figure E.3 shows these characteristics.

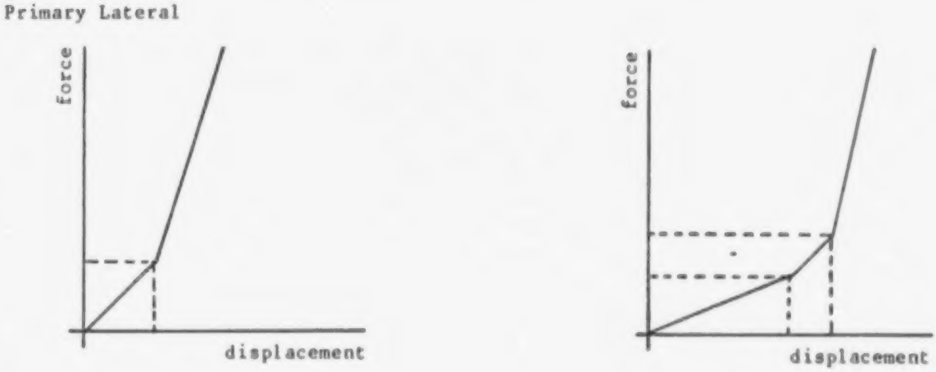


(a) without tractive effort

(b) with tractive effort

Figure E.3 Primary longitudinal suspension characteristics

The lateral suspension characteristics are shown in Figure E.4. The suspension is modelled as a bi-linear element, again representing lateral spring compression and a bump stop. However, the centre axles are given some extra lateral freedom, obtained by milling out the spring seats. This results in a range of relatively slack lateral axle motions. The model handles this by treating the centre axle lateral suspension as a tri-linear spring.



(a) outboard axles

(b) centre axles

Figure E.4 Primary lateral suspension characteristics

The vertical primary is modelled as a linear component.

(b) **Secondary suspension**. The secondary suspension is modelled in two parts. The lateral and vertical components are assumed to be linear and are placed at the level of the truck-to-bolster connection. The yaw suspension is assumed to be a friction-limited torsional stiffness at the centre-plate. This characteristic is seen in Figure E.5.

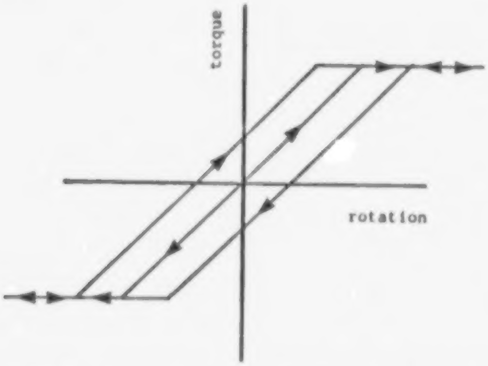


Figure E.5 Secondary yaw suspension characteristic

E.6 SIX-AXLE LOCOMOTIVE WITH STIFF CENTRE AXLES

This six-axle locomotive model is identical to the standard six-axle model, except that the extra lateral freedom associated with the centre axles (see Figure E.4) is removed. In this model, all the axles have identical primary lateral suspensions.

E.7 LOCOMOTIVES WITH INTER-CONNECTED TRUCKS

Both the six- and four-axle locomotives were also modelled with inter-truck connections. For these cars, the basic vehicle model remained unchanged, except for the addition of a lateral suspension link between the trucks. Figure E.6 shows how this arrangement could be set up on a six-axle locomotive. The net result is a coupling of the lead and trailing truck yaw motions. Simulation has shown that, for the purpose of this study, the model is relatively insensitive to the suspension stiffness and damping used.

E.8 SIMULATION CONSTANTS

All simulation runs were done for vehicle speeds of 6.71 m/s (15 mph) on curves with balanced superelevation and transitions. All vehicles were equipped with 40 inch diameter wheels with AAR 1:20 conical tread profiles. Rail profiles were for 136 lb RE rail.

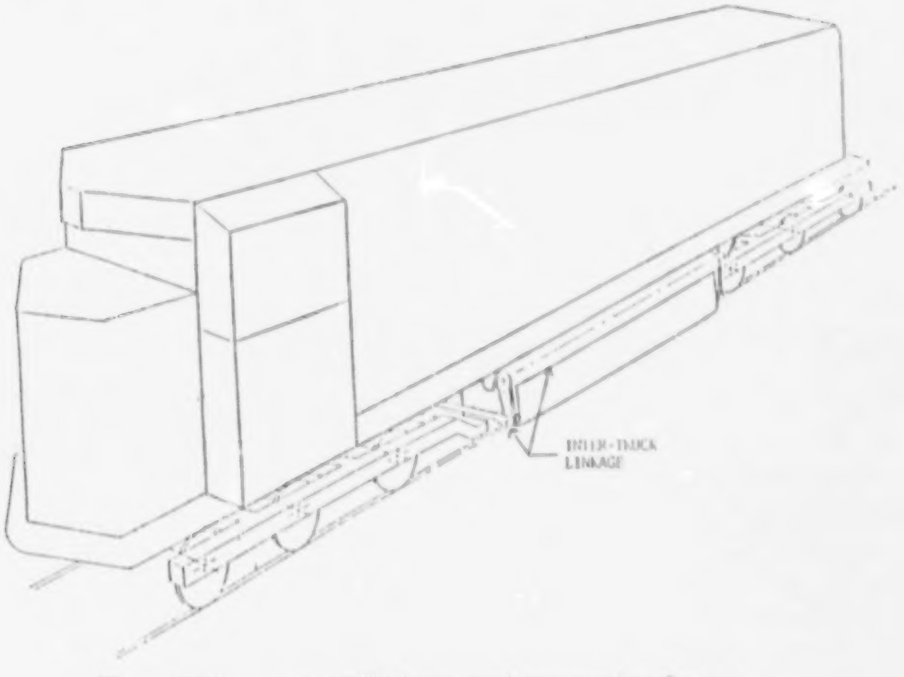


Figure E.6 A possible inter-truck connection for a diesel-electric locomotive

E.8 REFERENCES

1. Anderson, R.J., "Implementation of Multi-Body Dynamic Analysis on a Smaller Computer," *Proceedings of the 4th International ASME Computers in Engineering Conference*, Las Vegas, U.S.A., August 1984, pp. 721-726.

2. Anderson, R.J., *A'GEM -- Automatic Generation of Equations of Motion: User's Manual*, Queen's University Department of Mechanical Engineering, Dynamics Laboratory, Report DL/86/3, 1986.

3. Heller, R., and Cooperrider, N.K., *User's Manual for Asymmetric Wheel/Rail Contact Characterization Program*, U.S. Federal Railroad Administration Report FRD/ORD-78/05, December 1977.

4. Kalker, J.J., "A Fast Algorithm for the Simplified Theory of Rolling Contact," *Vehicle System Dynamics*, Vol. 11, pp. 1-13, 1982.

5. Garg, V.K., et al., *Development of a Simulation Methodology for Optimizing Locomotive Suspension Systems*, ASME Publication 77-RT-6.

APPENDIX F

LOCOMOTIVE CURVING SIMULATION RESULTS

The graphs and tables on the following pages are a representative sampling of the locomotive curving simulations run using the RACES program. Unless otherwise specified, all simulations were done with:

- vehicle speed of 15 mph
- AAR 1:20 conical wheels
- 136 lb rail
- superelevation set for a balance speed of 15 mph
- transition length set according to AREA guidelines
- half Kalber wheel/rail creep coefficients
- 2.0 metres of tangent track leading into the spiral transition
- no drawbar force.

Table F.1 shows the simulation parameters that were set as a function of track curvature.

TABLE F.1

CURVATURE-DEPENDANT SIMULATION PARAMETERS

Degree of Curvature (deg.)	Radius of Curvature (m)	Superelevation (rad.)	Transition Length (m)
2	873.25	.0056	5.95
4	436.63	.0112	11.91
6	291.08	.0167	17.86
8	218.31	.0223	23.81

All simulations with tractive effort were done by varying the locomotive tractive effort as it ran through the curve. It was seen that after a tractive effort change, the system adjusted quickly to a new steady state. Tests showed that the steady-state results were the same as if the entire simulation had been done at that tractive effort. Table F.2 shows the transition points that were used for switching tractive effort values, as well as the corresponding coefficients of friction used.

TABLE F.2

TRACTIVE-EFFORT-DEPENDANT SIMULATION PARAMETERS

Distance (m)	Tractive Effort (%)	Tread Coefficient of Friction	Flange Coefficient of Friction
0-50	18	.24	.10
50-75	24	.28	.10
75-100	30	.36	.10
100-125	36	.44	.10

Figure F.1 shows a complete run for the four-axle locomotive with tractive effort, and is included to illustrate the adjustment from one tractive effort to the next during a run. As these transitions are fairly quick and uninteresting, Figures F.2 through F.12 show the 18 percent tractive effort portion of each run, which shows locomotive behaviour during curve entry.

Each plot represents 50 meters of vehicle travel along the track. As noted before, each run starts on tangent track, proceeds through a spiral transition and moves into a constant radius circular curve. The distance coordinate represents the distance of the lead wheelset of the vehicle from the start of the run. Wheelsets are numbered sequentially from the front to the back of the vehicle. Table F.3 shows the placement of break points for each track curvaRture.

TABLE F.3

TRACK BREAK POINTS

Degree of Curvature (deg.)	Start of Tangent Track (m)	Start of Transition Section (m)	Start of Circular Curve (m)
2	0.0	2.0	7.95
4	0.0	2.0	13.91
6	0.0	2.0	19.86
8	0.0	2.0	25.81

The complete set of steady-state results are shown in Tables F.4 through F.8.

TABLE F.4

STANDARD SIX-AXLE LOCOMOTIVE CURVING WITH TRACTIVE EFFORT

		Flange Forces				Angles of Attack			
		Tractive Effort (%)				Tractive Effort (%)			
Degree of curve (deg)	Axle no.	18 Force (kN)	24 Force (kN)	30 Force (kN)	36 Force (kN)	18 Angle (deg)	24 Angle (deg)	30 Angle (deg)	36 Angle (deg)
2	1	48.	35.	23.	26.	-.30	-.30	-.30	-.30
	2	25.	22.	21.	24.	-.15	-.15	-.15	-.15
	3	0.	0.	0.	0.	-.00	-.00	-.00	-.00
	4	38.	29.	23.	29.	-.30	-.30	-.30	-.30
	5	19.	16.	16.	18.	-.15	-.15	-.15	-.15
	6	0.	0.	0.	0.	.00	.00	.00	.00
	Avg HR Force	22.	17.	14.	16.				
4	1	58.	44.	40.	32.	-.57	-.57	-.57	-.57
	2	35.	31.	35.	38.	-.30	-.30	-.30	-.30
	3	0.	0.	0.	0.	-.03	-.03	-.03	-.03
	4	40.	29.	30.	29.	-.50	-.50	-.50	-.50
	5	30.	27.	30.	32.	-.23	-.23	-.23	-.23
	6	0.	0.	0.	0.	.05	.05	.05	.05
	Avg HR Force	27.	22.	23.	22.				
6	1	63.	49.	44.	45.	-.73	-.73	-.73	-.73
	2	34.	53.	34.	41.	-.33	-.33	-.33	-.33
	3	-26.	-26.	-27.	-29.	.08	.08	.08	.08
	4	39.	26.	28.	34.	-.63	-.63	-.63	-.63
	5	28.	25.	23.	32.	-.23	-.23	-.23	-.23
	6	-19.	-18.	-21.	-24.	.17	.17	.17	.17
	Avg HR Force	27.	26.	22.	25.				
8	1	54.	38.	37.	28.	-.84	-.84	-.84	-.84
	2	30.	25.	30.	33.	-.30	-.30	-.30	-.30
	3	-39.	-38.	-42.	-46.	.26	.26	.26	.26
	4	54.	40.	40.	38.	-.84	-.84	-.84	-.84
	5	29.	26.	30.	33.	-.27	-.27	-.27	-.27
	6	-39.	-38.	-43.	-48.	.25	.25	.25	.25
	Avg HR Force	26.	22.	23.	22.				

TABLE F.5

INTERCONNECTED SIX-AXLE LOCOMOTIVE CURVING WITH TRACTIVE EFFORT

Degree of curve (deg)	Axle no.	Flange Forces				Angles of Attack			
		Tractive Effort (%)				Tractive Effort (%)			
		18	24	30	36	18	24	30	36
		Force (kN)	Force (kN)	Force (kN)	Force (kN)	Angle (deg)	Angle (deg)	Angle (deg)	Angle (deg)
2	1	NA	NA	NA	NA	NA	NA	NA	NA
	2								
	3								
	4								
	5								
	6								
	Avg HR Force								
4	1	33.	25.	18.	18.	-.34	-.42	-.46	-.43
	2	18.	20.	24.	20.	-.10	-.15	-.19	-.15
	3	0.	0.	0.	0.	.16	.14	.10	.13
	4	0.	0.	0.	0.	-.38	-.31	-.27	-.33
	5	0.	0.	-5.	0.	-.09	-.03	.02	-.04
	6	0.	0.	-31.	-24.	.18	.24	.28	.23
	Avg HR Force	9.	8.	7.	6.				
6	1	NA	NA	NA	NA	NA	NA	NA	NA
	2								
	3								
	4								
	5								
	6								
	Avg HR Force								
8	1	24.	18.	12.	15.	-.65	-.73	-.73	-.72
	2	17.	21.	21.	21.	-.10	-.20	-.20	-.18
	3	0.	-18.	-21.	-27.	.46	.37	.37	.38
	4	0.	0.	0.	0.	-.69	-.57	-.55	-.60
	5	17.	0.	-7.	0.	-.12	.00	.02	-.03
	6	-19.	-33.	-44.	-51.	.42	.53	.55	.50
	Avg HR Force	10.	7.	6.	6.				

TABLE F.6

STANDARD FOUR-AXLE LOCOMOTIVE CURVING WITH TRACTIVE EFFORT

Degree of curve (deg)	Axle no.	Flange Forces				Angles of Attack			
		Tractive Effort (%)				Tractive Effort (%)			
		18	24	30	36	18	24	30	36
		Force (kN)	Force (kN)	Force (kN)	Force (kN)	Angle (deg)	Angle (deg)	Angle (deg)	Angle (deg)
2	1	42.	28.	16.	12.	-.22	-.22	-.22	-.22
	2	0.	0.	0.	0.	-.03	-.03	-.03	-.03
	3	25.	16.	13.	13.	-.18	-.18	-.18	-.18
	4	0.	0.	0.	0.	.02	.02	.02	.02
	Avg HR Force	17.	11.	7.	6.				
4	1	58.	36.	36.	30.	-.44	-.44	-.44	-.44
	2	0.	0.	0.	0.	-.08	-.08	-.08	-.08
	3	33.	14.	19.	20.	-.33	-.33	-.33	-.33
	4	0.	0.	0.	0.	.04	.04	.04	.04
	Avg HR Force	23.	13.	14.	13.				
6	1	54.	39.	32.	27.	-.58	-.58	-.58	-.58
	2	0.	0.	0.	0.	-.05	-.05	-.05	-.05
	3	31.	26.	30.	29.	.55	.55	.55	.55
	4	0.	0.	0.	0.	-.00	-.00	-.00	-.00
	Avg HR Force	21.	16.	16.	14.				
8	1	59.	35.	36.	31.	-.81	-.81	-.81	-.81
	2	0.	0.	0.	0.	-.09	-.09	-.09	-.09
	3	32.	27.	31.	31.	-.73	-.73	-.73	-.73
	4	0.	0.	0.	0.	.00	.00	.00	.00
	Avg HR Force	23.	16.	17.	16.				

TABLE F.7

STANDARD FOUR-AXLE LOCOMOTIVE CURVING WITH TRACTIVE EFFORT

Degree of curve (deg)	Axle no.	Flange Forces				Angles of Attack			
		Tractive Effort (%)				Tractive Effort (%)			
		18	24	30	36	18	24	30	36
		Force (kN)	Force (kN)	Force (kN)	Force (kN)	Angle (deg)	Angle (deg)	Angle (deg)	Angle (deg)
2	1	26.	21.	13.	10.	-.15	-.15	-.15	-.15
	2	0.	0.	0.	0.	.03	.03	.03	.03
	3	0.	0.	0.	0.	-.12	-.12	-.12	-.12
	4	0.	0.	0.	0.	.07	.07	.07	.07
	Avg HR Force	7.	5.	3.	3.				
4	1	28.	18.	10.	16.	-.26	-.31	-.31	-.28
	2	0.	0.	0.	0.	.11	.05	.05	.08
	3	0.	0.	0.	0.	-.27	-.25	-.23	-.28
	4	0.	-20.	-16.	0.	.10	.17	.13	.08
	Avg HR Force	7.	5.	3.	4.				
6	1	39.	26.	26.	NA	-.48	-.53	-.51	NA
	2	0.	0.	0.		.08	.00	.03	
	3	0.	0.	0.		-.29	-.19	-.23	
	4	-19.	-36.	-40.		.25	.35	.32	
	Avg HR Force	10.	7.	7.					
8	1	27.	14.	17.	15.	-.57	-.64	-.63	-.63
	2	0.	0.	0.	0.	.17	.08	.10	.10
	3	0.	0.	0.	0.	-.57	-.30	-.33	-.34
	4	-26.	-40.	-47.	-54.	.33	.43	.40	.38
	Avg HR Force	7.	4.	4.	4.				

TABLE F.8

FREIGHT CAR AND LOCOMOTIVE CURVING WITHOUT TRACTIVE EFFORT

Deg of curve (deg)	Axl no.	100 ton Freight		Standard 4 Axle Locomotive		Intercon. 4 Axle Locomotive		Standard 6 Axle Locomotive		Intercon. 6 Axle Locomotive	
		Force (kN)	Angle (deg)	Force (kN)	Angle (deg)	Force (kN)	Angle (deg)	Force (kN)	Angle (deg)	Force (kN)	Angle (deg)
2	1			60.	-.20	40.	-.15	71.	-.27	43.	-.19
	2			0.	-.02	0.	.03	36.	-.14	15.	-.05
	3			41.	-.16	29.	-.13	0.	.0	0.	.08
	4	NA	NA	0.	.02	0.	.05	59.	-.25	31.	-.20
	5							36.	-.11	15.	-.06
	6							0.	.02	0.	.07
	Avg HR Frc			25.		17.		34.		17.	
4	1	96.	-.24	100.	-.42	68.	-.30	105.	-.57	70.	-.39
	2	20.	-.04	0.	-.06	0.	.06	60.	-.30	29.	-.12
	3	96.	-.24	72.	-.34	46.	-.27	0.	-.03	0.	.16
	4	20.	-.04	0.	.02	0.	.08	88.	-.53	45.	-.39
	5							50.	-.24	20.	-.10
	6							0.	.03	-27.	.16
	Avg HR Frc	58.		43.		29.		51.		27.	
6	1			115.	-.61	77.	-.42	117.	-.74	82.	-.57
	2			0.	-.08	0.	.11	63.	-.35	38.	-.19
	3			89.	-.53	58.	-.42	-31.	.07	-36.	.24
	4	NA	NA	0.	.01	0.	.11	93.	-.67	50.	-.50
	5							47.	-.25	0.	-.08
	6							-30.	.15	-38.	.31
	Avg HR Frc			51.		34.		53.		28.	
8	1	120.	-.52	121.	-.83	82.	-.57	123.	-.92	90.	-.73
	2	31.	-.13	0.	-.12	0.	.16	63.	-.38	39.	-.20
	3	120.	-.52	99.	-.73	59.	-.50	-61.	.18	-41.	.37
	4	31.	-.13	0.	-.02	-41.	.23	98.	-.81	52.	-.63
	5							48.	-.26	0.	-.08
	6							-52.	.27	-39.	.45
	Avg HR Frc	76.		55.		35.		55.		30.	

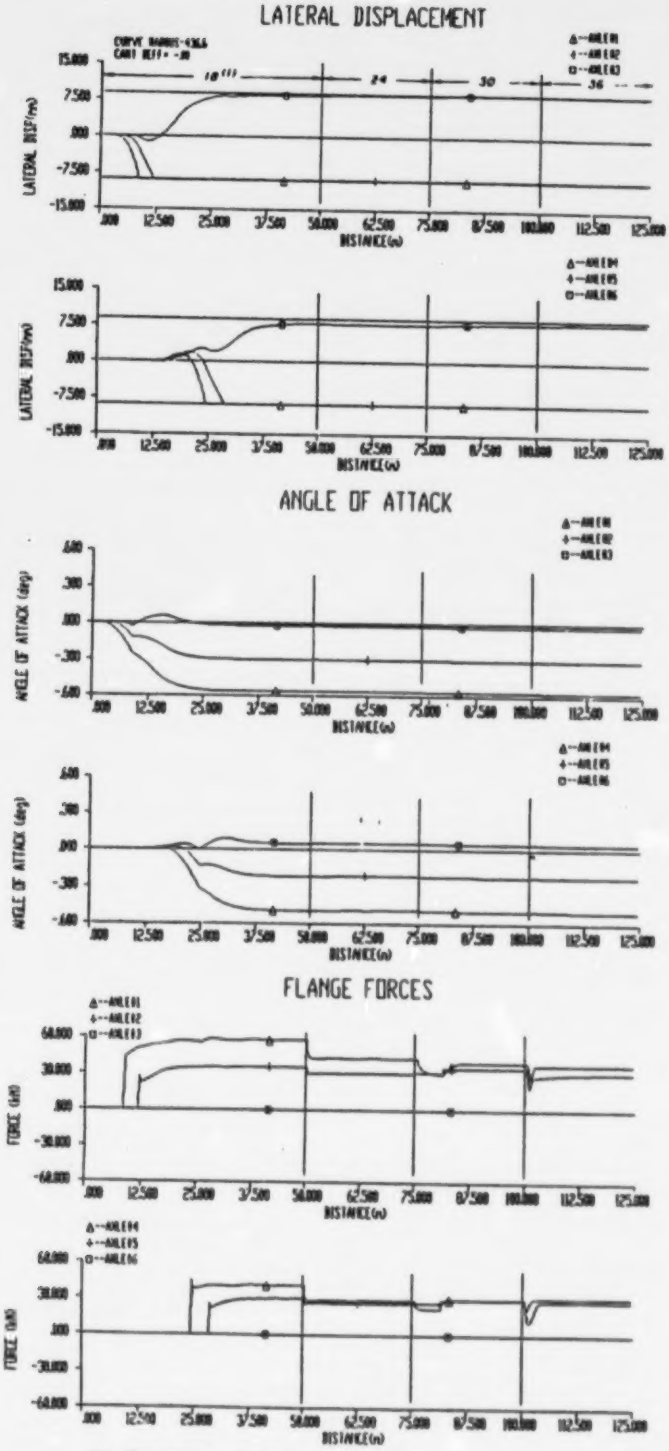
The plots on the following pages show the simulated locomotive curving performances for the four configurations of locomotive tested. These are:

- a six-axle locomotive
- a four axle locomotive
- a six-axle locomotive with interconnected trucks
- a four-axle locomotive with interconnected trucks.

A loaded 100-ton hopper car was also simulated for comparative purposes.

The results shown for each case are:

- lateral displacements for each wheelset
- angle of attack for each wheelset
- flange force for each wheelset, if flanging occurs.



(1) Percentage Tractive Efforts

Figure F.1 Six-axle locomotive, 4° degree curve

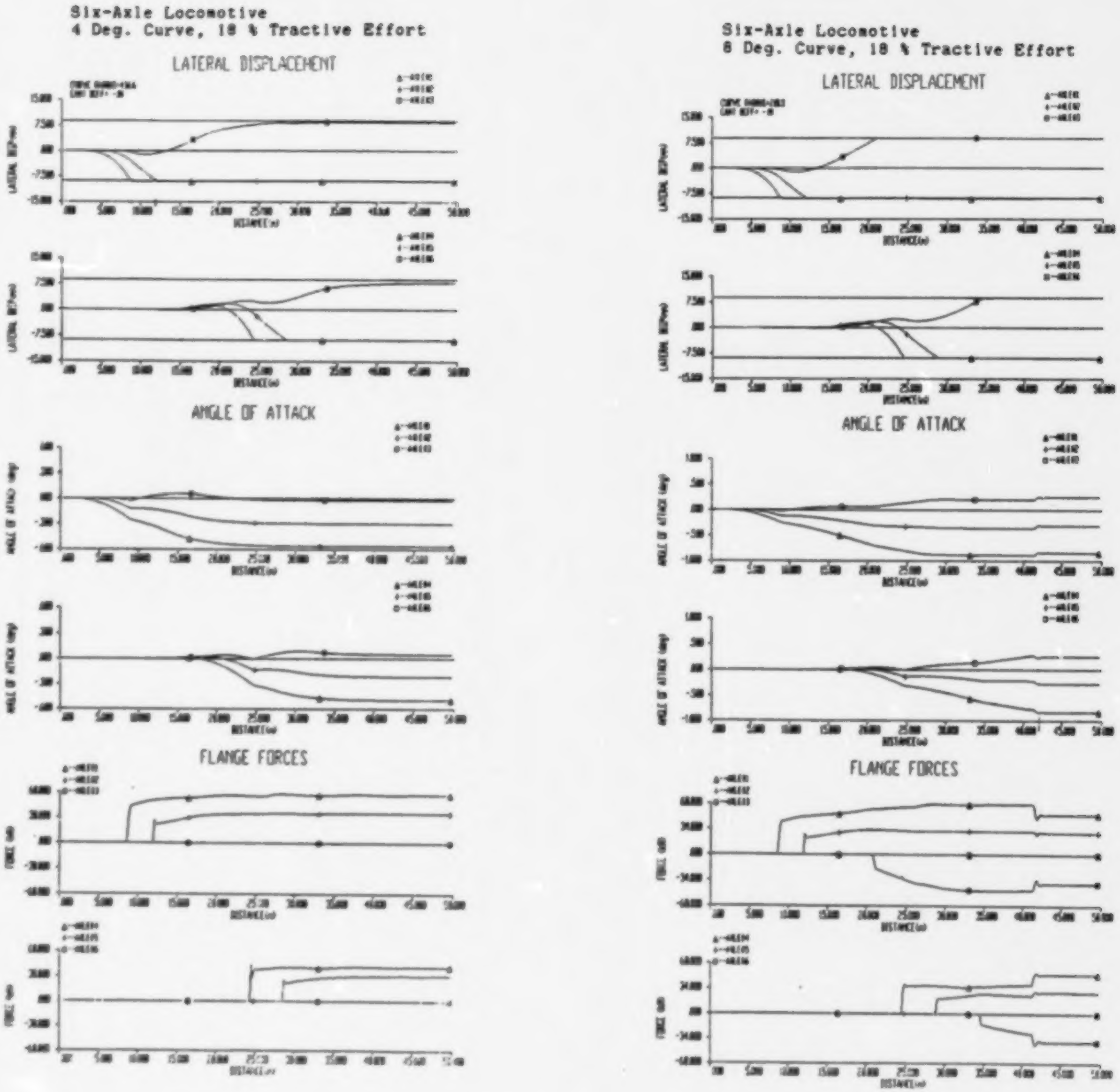
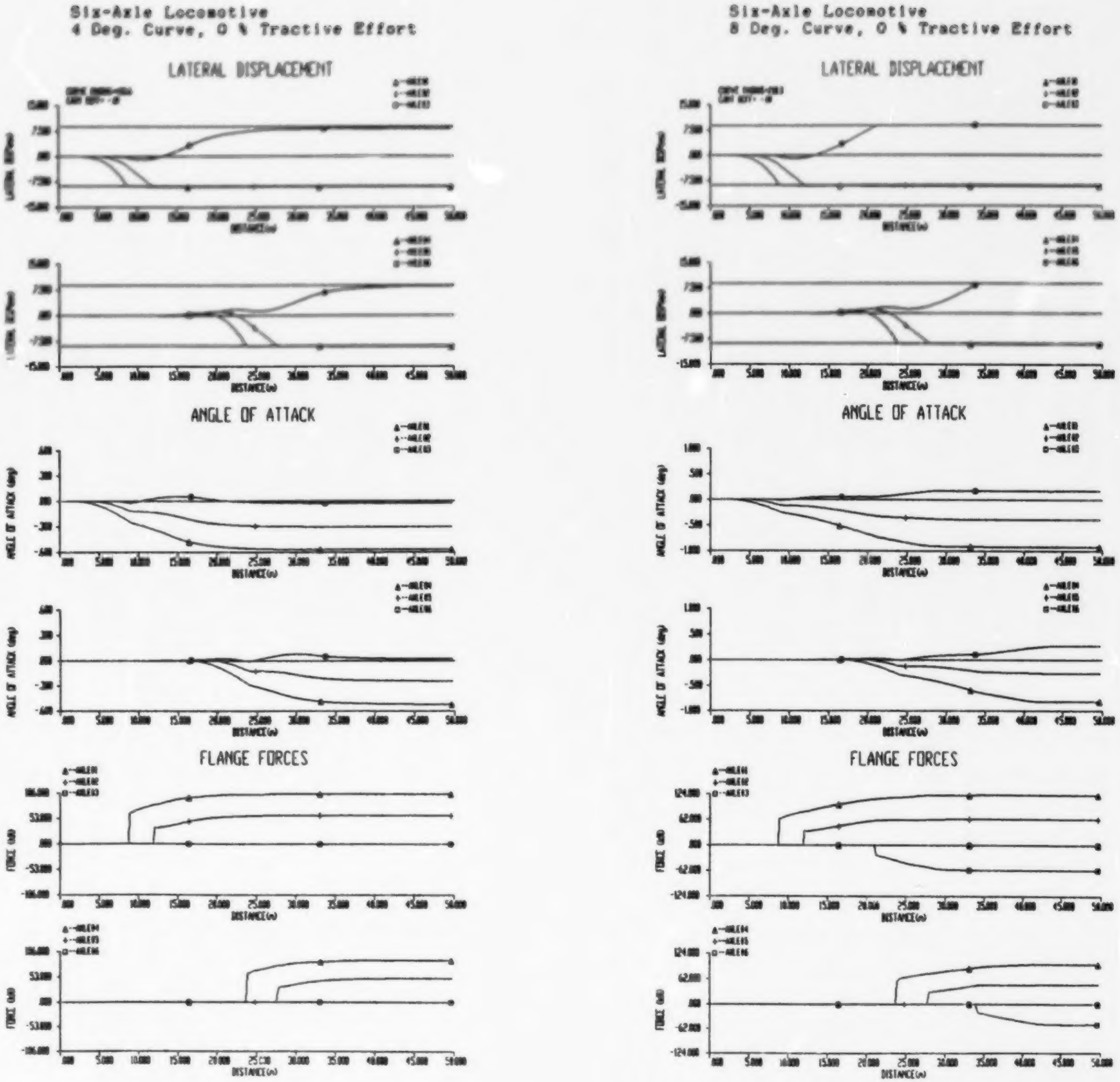


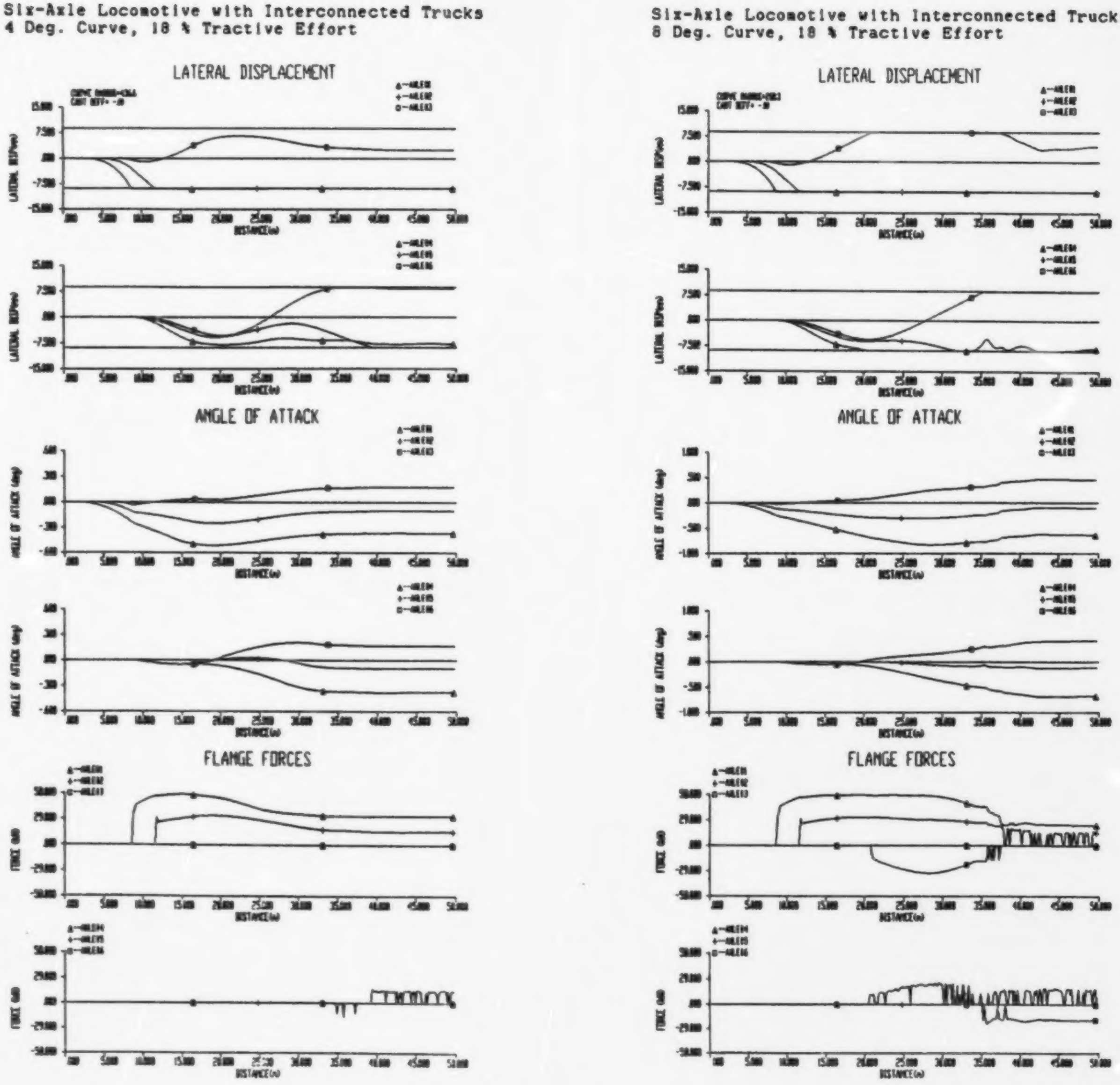
Figure F.2 Six-axle locomotive with tractive effort



Six-Axle Locomotive
4 Deg. Curve, 0 % Tractive Effort

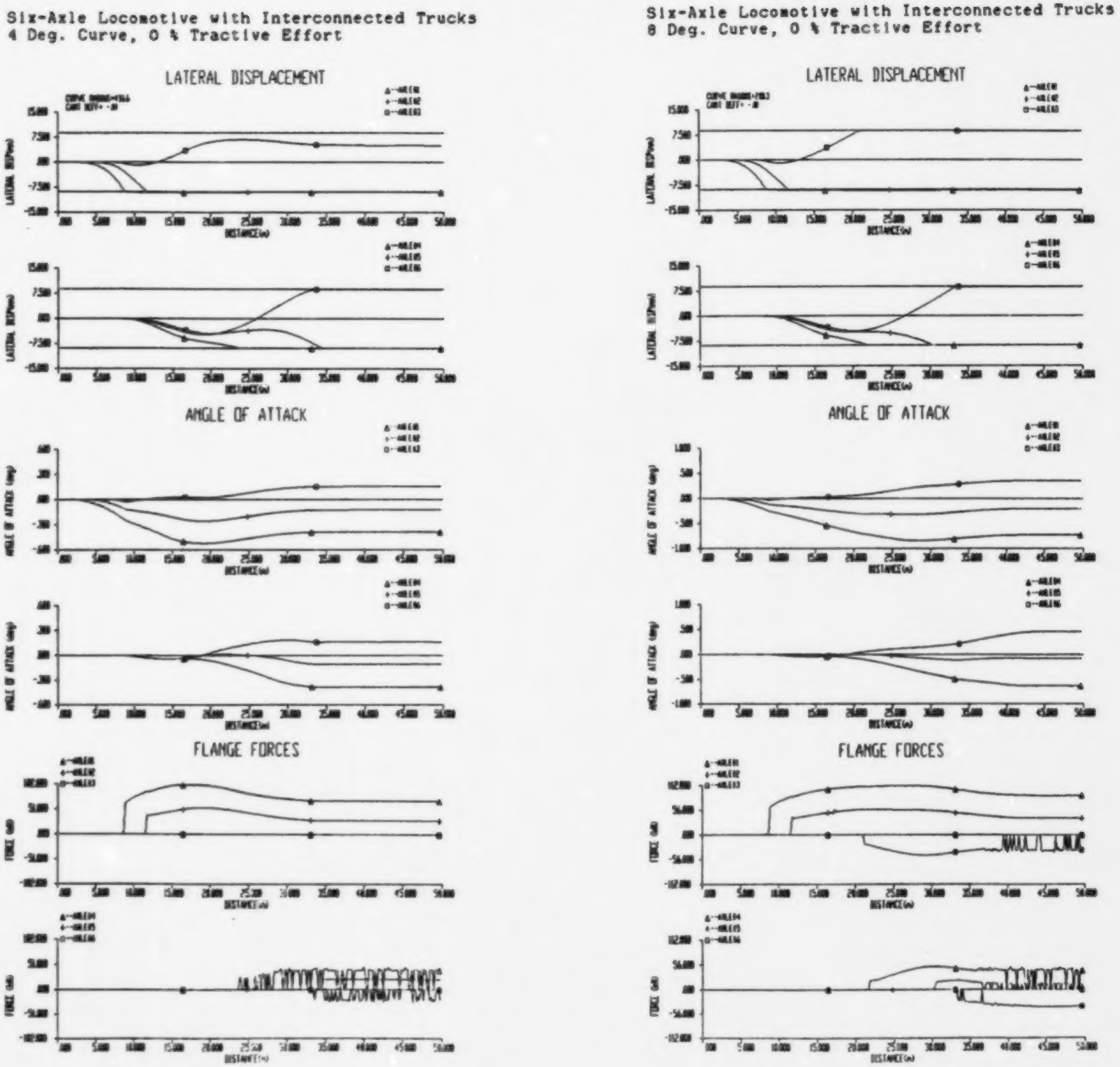
Six-Axle Locomotive
8 Deg. Curve, 0 % Tractive Effort

Figure F.3 Six-axle locomotive without tractive effort


Six-Axle Locomotive with Interconnected Trucks
4 Deg. Curve, 18 % Tractive Effort

Six-Axle Locomotive with Interconnected Trucks
8 Deg. Curve, 18 % Tractive Effort

Figure F.4 Interconnected six-axle locomotive with tractive effort



Six-Axle Locomotive with Interconnected Trucks
4 Deg. Curve, 0 % Tractive Effort

Six-Axle Locomotive with Interconnected Trucks
8 Deg. Curve, 0 % Tractive Effort

Figure F.5 Interconnected six-axle locomotive without tractive effort

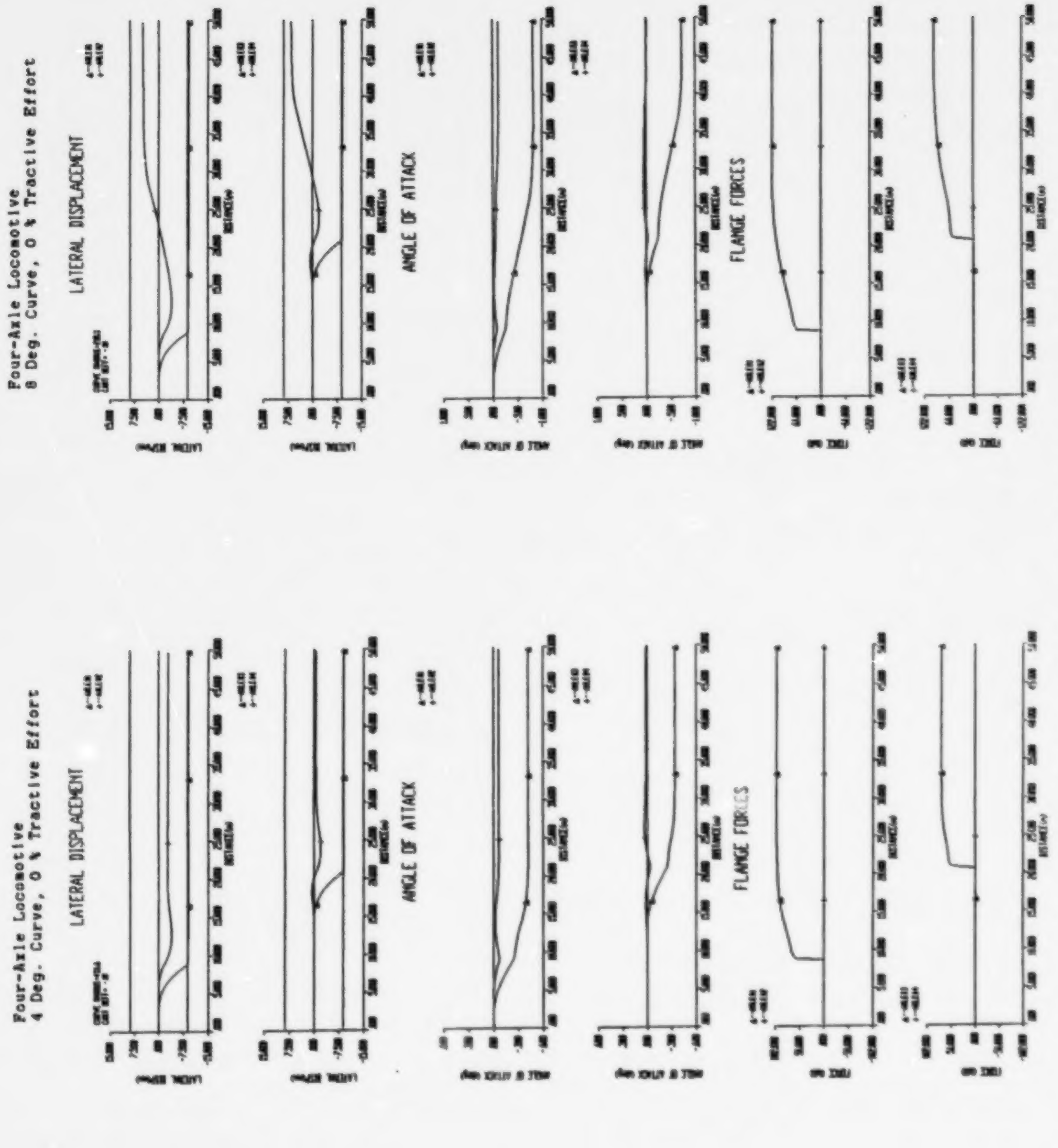


Figure F.6 Four-axle locomotive with tractive effort

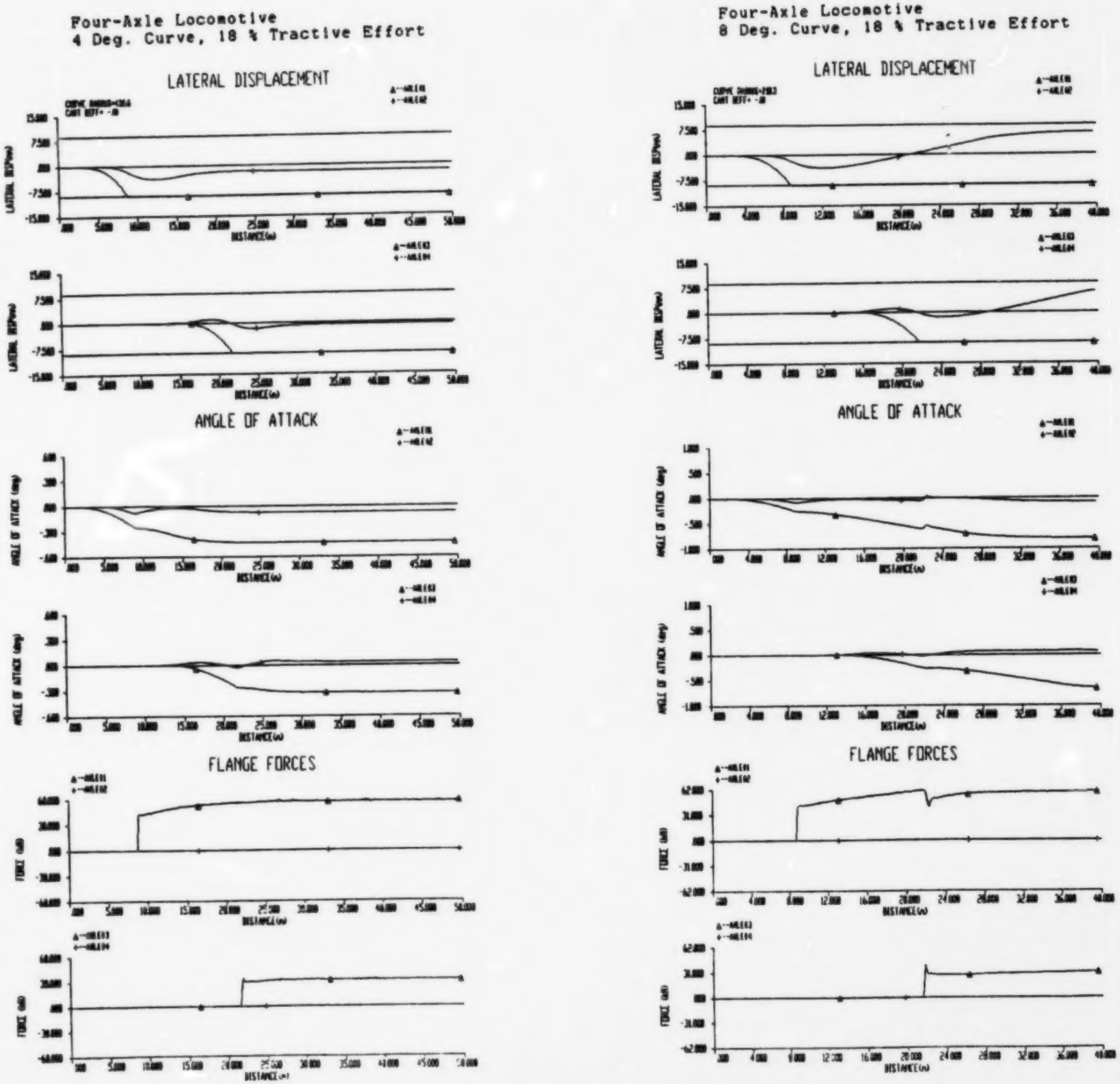


Figure F.7 Four-axle locomotive without tractive effort

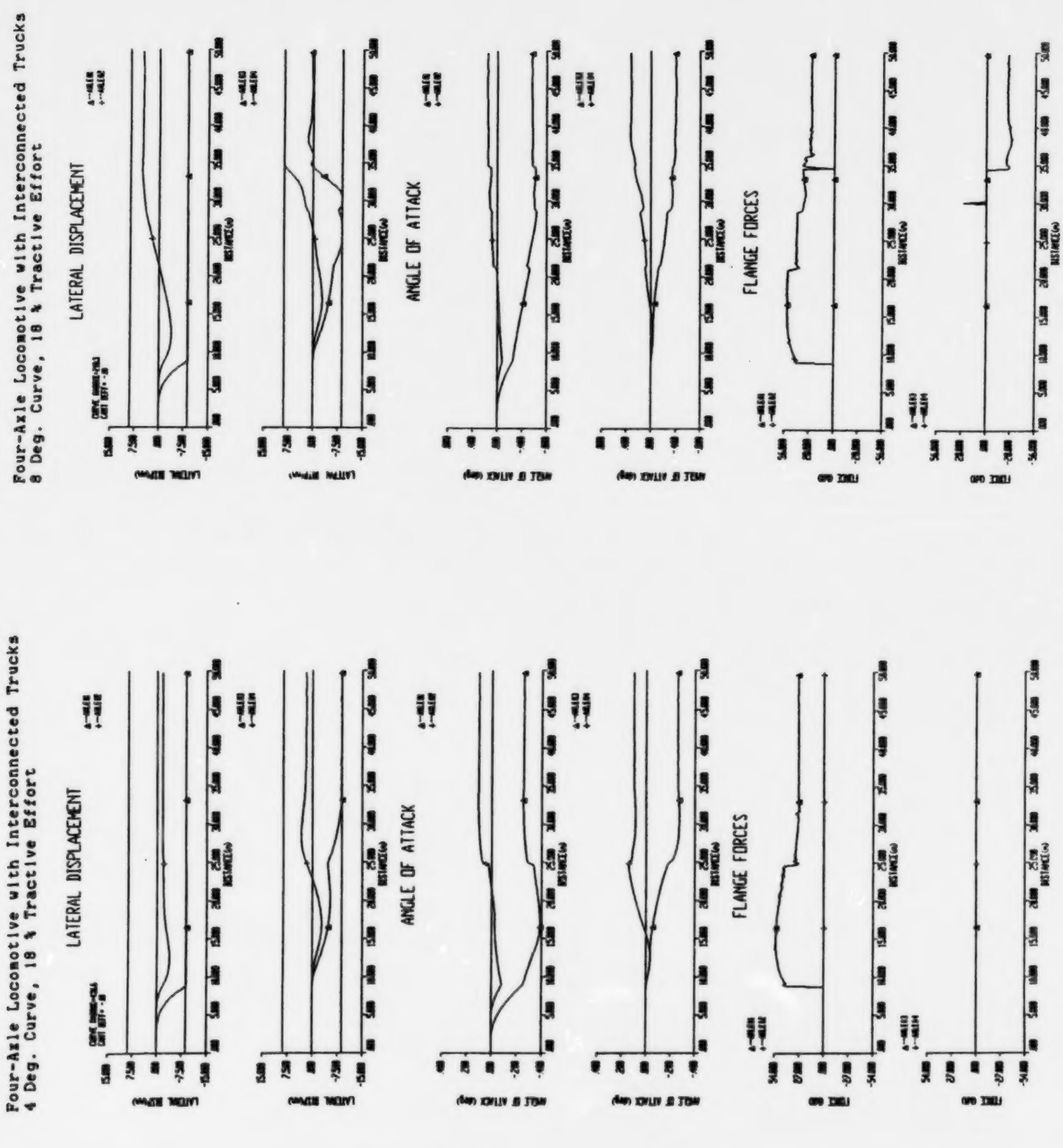
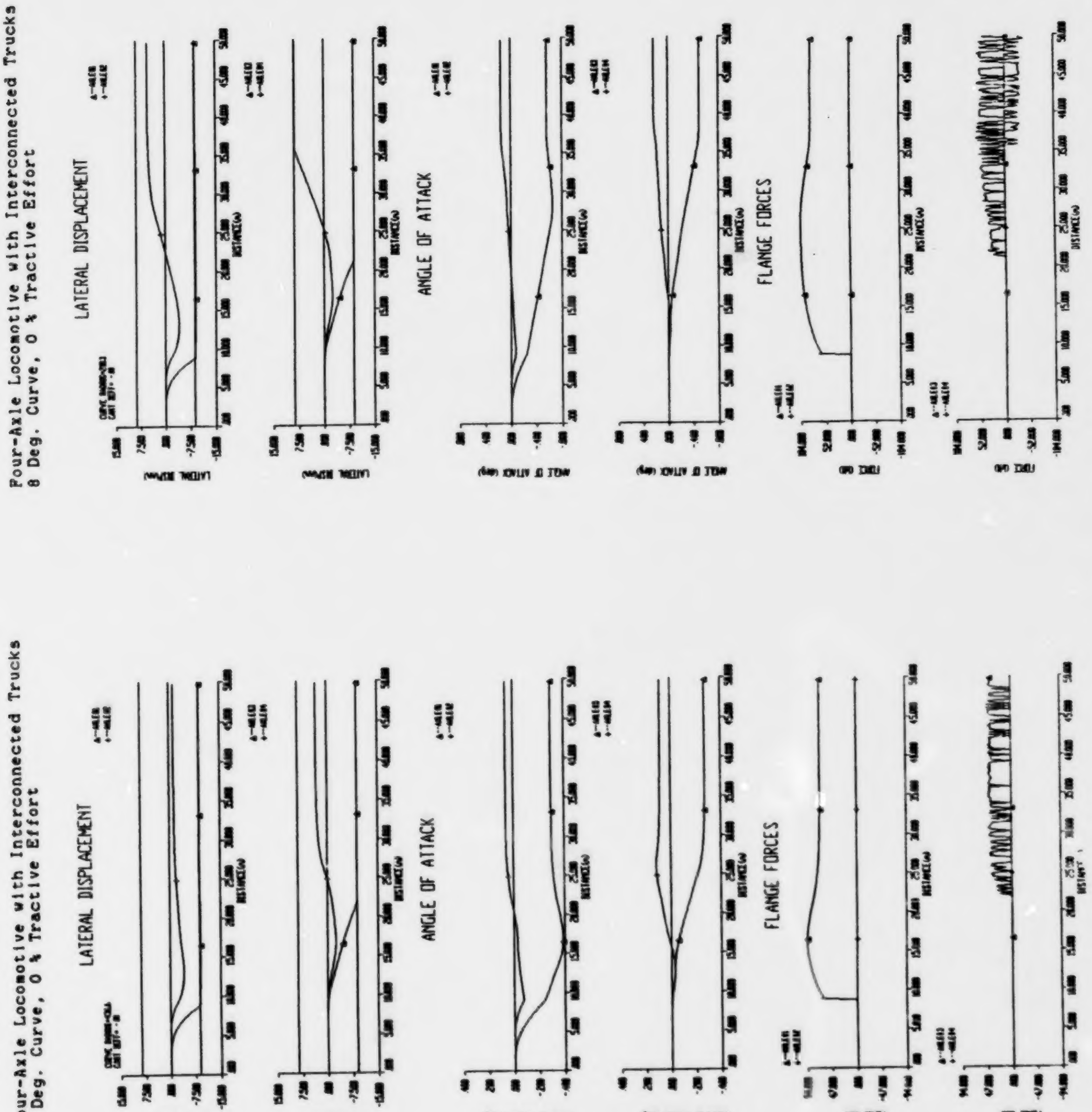


Figure F.8 Interconnected four-axle locomotive with tractive effort

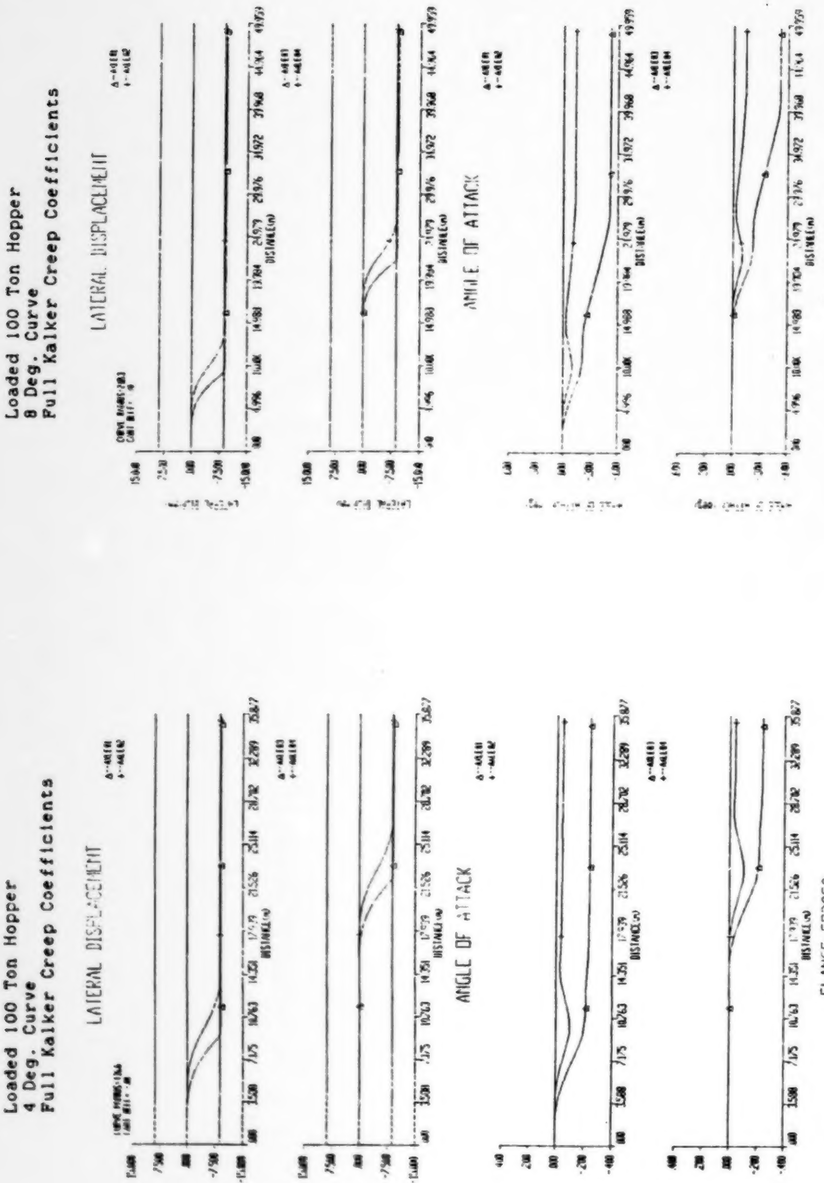
Figure F.9 Interconnected four-axle locomotive without tractive effort



Four-Axle Locomotive with Interconnected Trucks
8 Deg. Curve, 0 % Tractive Effort

Four-Axle Locomotive with Interconnected Trucks
4 Deg. Curve, 0 % Tractive Effort

Figure F.9 Interconnected four-axle locomotive without tractive effort



Loaded 100 Ton Hopper
8 Deg. Curve
Full Kalker Creep Coefficients

Loaded 100 Ton Hopper
4 Deg. Curve
Full Kalker Creep Coefficients

Figure F.10 Loaded 100-ton hopper car

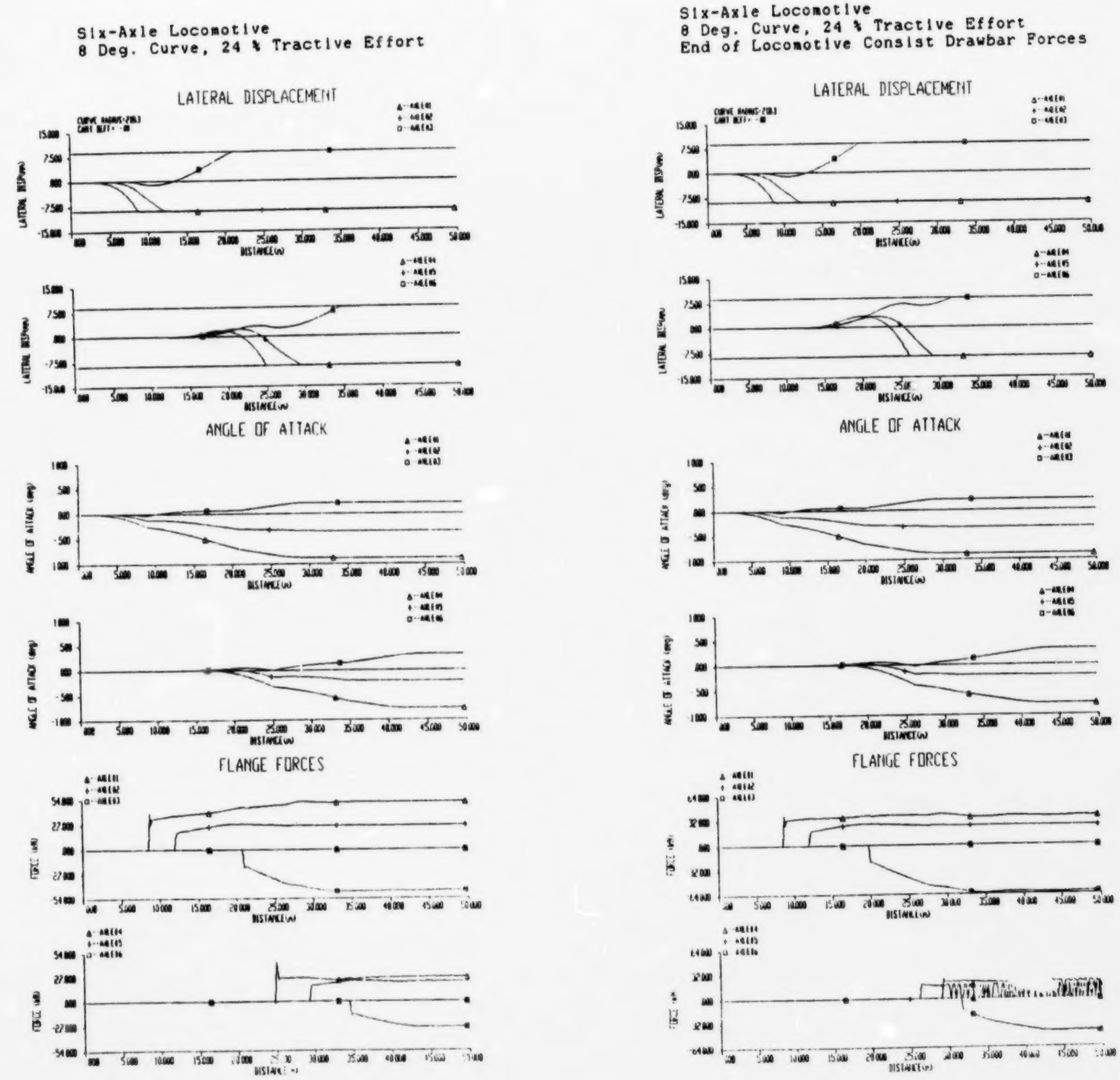
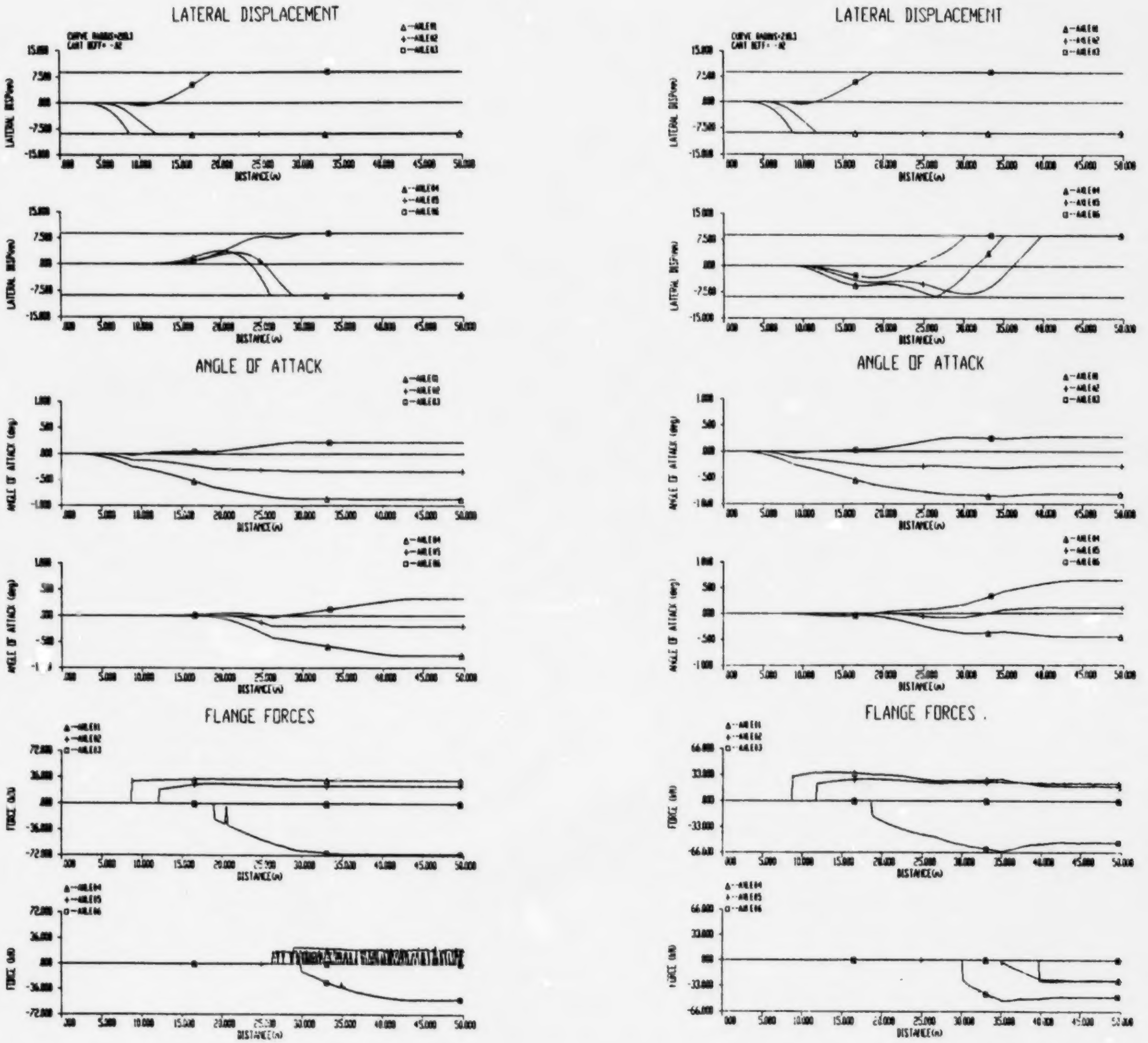


Figure F.11 Six-axle locomotive with drawbar forces



Six-Axle Locomotive
8 Deg. Curve, 24 % Tractive Effort, 7 mph
End of Locomotive Consist Drawbar Forces

Six-Axle Locomotive with Interconnected Trucks
8 Deg. Curve, 24 % Tractive Effort, 7 mph
End of Locomotive Consist Drawbar Forces

Figure F.12 Six-axle locomotive, underspeed, with drawbar forces

APPENDIX G

UNIT SIZE AND SPECIALIZATION IMPACTS ON LOCOMOTIVE UTILIZATION

A time-based simulation model was developed in GPSS to depict a major locomotive shop with trains arriving and departing on a fixed schedule. The model assumes that there are four train types: coal, bulk, freight and express. For each train type, the total train weight is based on a normal distribution with a given mean weight and standard deviation of weight. A given horsepower-per-ton value is used to calculate the total horsepower requirements for each type of train. The weights and horsepower-per-ton values differ between arrivals and departures for the same train type. The model was run for a 15-day period.

The model has two logic paths, one for arrivals and the other for departures. The arrivals are generated at a fixed time each day for the duration of the simulation. After the total weight and horsepower of the train is determined, the model calculates the number of locomotives required. A separate record is created for each locomotive to accumulate various time statistics while the locomotive is moving through the shop. Upon arrival, the locomotive is transferred to an inspection queue with a fixed time delay. The locomotives are examined by a fixed number of inspectors. The time taken for this examination is based on a discrete probability distribution.

The locomotives are then transferred to the service pool or to the maintenance queue. The distribution between service pool and maintenance queue is a fixed proportion. The locomotives in the maintenance queue are split into two groups, for scheduled and unscheduled work. This division is also based on a fixed percentage. For each type of work, the time taken is based on a discrete probability function. The locomotive is removed from the maintenance queue to the work bay on a priority basis inversely related to the work time. There is a fixed number of work bays, with the possibility of one of the work bays being shut down during the day. If a work bay is shut down while a locomotive is being worked on, the locomotive remains in the work bay until the bay reopens. After the locomotive is finished in the work bay, it is released to the service pool with a fixed time delay.

Departures are generated at fixed times each day for the duration of the simulation. The call for marshalling a train occurs a fixed time period before the scheduled departure. At this time, the total train weight and horsepower are calculated. The total horsepower required is decreased as a locomotive is added from the service pool. If the total horsepower is below a fixed percentage of the locomotive horsepower, the model will drop a calculated number of cars to reduce the horsepower required to zero. The train is then set for departure. If there are no locomotives in the service pool, the train waits for a fixed time period before checking the service pool again.

When the train has the required locomotives, it will wait a fixed time for marshalling. The model will compare current time to that of the train's scheduled departure. If the train is ready before departure, it waits until departure time. If the train is late, the model accumulates the delay time.

The first model run was the base case using 3,000 hp locomotives. It is called "BASE CASE". The various fixed values for the base case are given in Table G.1.

TABLE G.1
BASE CASE

	Train Type	Mean Weight	Standard Deviation	Horsepower per Ton
Arrival	Coal	3,000	0	4.5
	Bulk	3,500	500	1.5
	Freight	4,500	700	2.5
	Express	3,500	500	3.5
Depart	Coal	14,900	0	1.0
	Bulk	11,500	500	1.0
	Freight	6,000	850	1.3
	Express	5,000	500	2.0

Transfer Time to Inspection Queue: 180 minutes
Inspection Time:
No Work/Work Percentage: 88.3
Scheduled/Unscheduled Percentage: 38.2
Transfer Time to Service Pool: 60 minutes
Time Before Scheduled Departure: 120 minutes
Marshalling Time: 60 minutes

The second set of runs was to determine the model's sensitivity to locomotive size. There were two variations:

(a) Using a pool of two locomotives at 3,000 hp for every one locomotive at 6,000 hp. This model was called "25 CASE".

(b) Using a pool of 6,000 hp locomotives only. This model was called "FULL CASE".

For both runs, the total locomotive horsepower pool was equivalent to that of the base case. The inspectors and maintenance bays were decreased by the percentage decrease in the total starting locomotive pool. After checking the model results, revised runs were made. These revised runs increased the number of maintenance bays until the maintenance queue delays were equivalent to those of the base case.

The third set of runs was to determine the model's sensitivity to assigned locomotive service. There were two variations:

(a) Using dedicated locomotives for bulk and coal trains. These were 3,850 hp locomotives. The horsepower per ton is 0.66. If there were no 3,850 hp locomotives, then 3,000 hp locomotives were used at 1.0 horsepower per ton. This model was called "BULK CASE".

(b) Using dedicated locomotives for coal and bulk. These were 3,850 hp locomotives allocated at 0.66 horsepower per ton. There were also different dedicated locomotives for the express trains. These were 6,000 hp locomotives allocated at 2.0 horsepower per ton. If either the 3,850 hp or 6,000 hp locomotives were unavailable, 3,000 hp locomotives were used at 2.0/1.0/1.0 horsepower per ton for express, bulk, and coal trains, respectively. The 3,000 hp locomotives for freight were used at 1.3 horsepower per ton. This model was called "3 LOCOS".

For both variations the arriving trains had their horsepower per ton values changed to balance incoming locomotives to outgoing locomotives.

TABLE G.2

CASE SHP1-3A

	Train Type	Mean Weight	Standard Deviation	Horsepower per Ton
Arrive	Coal	3,000	0	2.56
	Bulk	3,500	500	2.2
	Freight	4,500	700	1.73
	Express	3,500	500	2.86
Depart	Coal	14,900	0	1.0
	Bulk	11,500	500	1.0
	Freight	6,000	850	1.3
	Express	5,000	500	2.0

Transfer Time to Inspection Queue:	180 Minutes
Inspection Time:	
No Work/Work Percentage:	88.3/11.7
Scheduled/Unscheduled Percentage:	38.2/61.8
Transfer Time to Service Pool:	60 Minutes
Time Before Scheduled Departure:	180 Minutes
Marshalling Time:	120 Minutes

TABLE G.3
CASE SHP1-3B

	Train Type	Mean Weight	Standard Deviation	Horsepower per Ton
Arrive	Coal	3,000	0	4.5
	Bulk	3,500	500	3.29
	Freight	4,500	700	2.46
	Express	3,500	500	2.86
Depart	Coal	14,900	0	1.0
	Bulk	11,500	500	1.0
	Freight	6,000	850	1.3
	Express	5,000	500	2.0

Transfer Time to Inspection Queue:	180 Minutes
Inspection Time:	
No Work/Work Percentage:	88.3/11.7
Scheduled/Unscheduled Percentage:	38.2/61.8
Transfer Time to Service Pool:	60 Minutes
Time Before Scheduled Departure:	180 Minutes
Marshalling Time:	120 Minutes

TABLE G.4
CASE SHP1-3C

	Train Type	Mean Weight	Standard Deviation	Horsepower per Ton
Arrive	Coal	3,000	0	3.28
	Bulk	3,500	500	2.17
	Freight	4,500	700	1.73
	Express	3,500	500	2.86
Depart	Coal	14,900	0	1.0
	Bulk	11,500	500	1.0
	Freight	6,000	850	1.3
	Express	5,000	500	2.0

Transfer Time to Inspection Queue:	180 Minutes
Inspection Time:	
No Work/Work Percentage:	88.3/11.7
Scheduled/Unscheduled Percentage:	38.2/61.8
Transfer Time to Service Pool:	60 Minutes
Time Before Scheduled Departure:	180 Minutes
Marshalling Time:	120 Minutes

TABLE G.5
SENSITIVITY TO LOCOMOTIVE SIZE

		Base Case	25 Case	Full Case
Starting Locomotives:	Service Pool 1	15	10	
	Service Pool 2		5	8
	Inspection 1	12	6	
	Inspection 2		3	6
	Maintenance 1	3	2	
	Maintenance 2		1	2
		30	27	16
Inspection Bays		7	4	3
Maintenance Bays		5.00	4.75	2.67
Ending Locomotives:	Service Pool 1	3	0	
	Service Pool 2		0	1
Time in Service Pool:	Pool 1-Time (mins.)	135	409	
	Pool 1-Number	744	450	
	Pool 2-Time		90	271
	Pool 2-Number		148	403
Average Time (minutes)		135	330	271
Day 15 Status (queue sizes and times)				
Fixed Transfer		4	4	5
Queue Inspection		8	9	2
Avg. Time in Queue		28	46	78
Inspection		7	5	3
Queue Maintenance		8	6	5
Avg. Time in Queue		881	835	834
Maintenance		5	5	3
Transfer to Service Pool		3	1	3
		38	30	22
Train Delay	- Number of Trains	12	16	5
	- Average Minutes/Train	62	77	109
	- Total Minutes	744	1232	545
Cars Left		115	80	292
Car Delay (minutes)		918	460	1714
Total Equivalent Train Delay		1262	1592	1859
Locomotives into Maintenance		89	81	48
Total Number of Locomotives		742	592	404
Total Horsepower		2223	2238	2412
Total Avg. Locomotive Delay (minutes)		266	474	447

TABLE G.6

SENSITIVITY TO ASSIGNED SERVICE

		Base Case	Bulk Case	3 Locos Case
Starting Locomotives:	Service Pool 1	15	6	7
	Service Pool 2		7	7
	Service Pool 3			3
	Inspection 1	12	6	4
	Inspection 2		5	4
	Inspection 3			2
	Maintenance 1	3	1	1
	Maintenance 2		1	1
	Maintenance 3			1
		30	26	30
Inspection Bays		7	5	5
Maintenance Bays		5	4	4
Ending Locomotives:	Service Pool 1	3	0	0
	Service Pool 2		0	0
	Service Pool 3			5
Time in Service Pool:	Pool 1 - Time (mins)	135	327	458
	Pool 1 - Number	744	302	191
	Pool 2 - Time		337	317
	Pool 2 - Number		238	234
	Pool 3 - Time			1671
	Pool 3 - Number			64
Average Time		135.00	331.41	549.28
Day 15 Status (queue sizes and times)				
Fixed Transfer		4	6	4
Queue Inspection		8	4	1
Avg. Time in Queue		28	36	8
Inspection		7	5	5
Queue Maintenance		8	7	8
Avg. Time in Queue		881	645	684
Maintenance		5	4	4
Transfer to Service Pool		3	0	3
		38	26	30
Train Delay	- Number of Trains	12	12	17
	- Average Minutes/Train	62	103	167
	- Total	744	1236	2839
Cars Left		115	174	113
Car Delay (minutes)		918	1026	357
Total Equivalent Train Delay		1262	2019	3344
Locomotives into Maintenance		89	67	66
Total Locomotives		757	535	484
Total Horsepower		2223	1822	1828
Total Avg Locomotive Delay (minutes)		266	443	637

APPENDIX H

TECHNOLOGY ASSESSMENT:
POWERED FREIGHT RAILCARS

prepared for

Canadian Institute of Guided Ground Transport
Queen's University at Kingston, Ontario

re: CIGGT Project No. PRO-386

by

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SUMMARY

This report summarizes the results of an evaluation to determine the feasibility of adapting electrically powered rail trucks under 100-ton gondola and covered hopper freight railcars. Standard design configurations and dimensional limitations employed by CN and CP Rail and other major US railcar manufacturers for unit train operations were considered.

The study concludes that the adaptation of two General Motors GP-style locomotive trucks using standard 40-inch wheels and D77 traction motor and gearcase assemblies could be considered. Car structures would be redesigned with a raised centre-sill and offset couplers, and would exhibit air filtration/power control cabinets located at its ends. The powered trucks would be electrified through a multiple cable connector arrangement and be coupled to an adjacent locomotive in a semi-permanent married-pair configuration. The concepts described in this report would be suitable for bi-directional operation and utilize AAR standard equipment accepted by North American railway authorities.

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1. INTRODUCTION

In evaluating alternative methods for the improvement of high-productivity integral train (HPIT) operations for bulk commodities (e.g. coal, potash, grain, etc.), various designs of motive power units were considered. In the concepts proposed, electrically powered slave-type "booster" cars would eliminate the necessity for supplementary locomotives. Additional motive power units are often employed to provide the make-up power and tractive effort required for the movement of heavy-haul unit trains over specific grade crossings.

The design considered would allow a specially adapted 100-ton gondola or hopper car to operated in place of the make-up locomotive(s). In its loaded configuration, the unit would have a gross weight equivalent to that of a working locomotive (131 tons) and could be used for additional tractive effort for train movement. When dead-headed or when additional power is not required, the car's "light" weight would be considerably less than that of a substitute locomotive and could thus potentially provide some conservation to the train's overall energy and maintenance cost requirements.

2. OPERATIONAL REQUIREMENTS

2.1 Gondola Cars

In considering the adaptation of powered 100-ton gondola cars for rotary dump operations, specific design standards were required to be maintained. Gondola cars are widely used for unit coal or ore train service by major Canadian railways and have design features suited to the method by which the cars are emptied.

Dump sites used by Canadian railways generally employ a clamp and body-beam arrangement which is raised from track level and used to secure the loaded railcar onto its track. Car and track platforms are rotated together in the process through approximately 120° to unload the carried material. The mechanical unloading occurs with the car still attached to the train, each car having one fixed and one rotary coupler. Trains are assembled with cars matched fixed end to rotary end.

In order to maintain existing dump site unloading equipment, new car designs should have equivalent dimensions as to:

• overall height	12' 8"
• overall width	10' 8"
• length over coupling faces	53' 1"
• coupler centreline	2' 10"
• coupler type	'F'-type, one fixed and one rotary
• clearance	AAR plate 'C'

In order to gain greater acceptance by railway authorities, new car designs would be required to utilize as many standard railway car and truck components (assemblies) as possible. Rail trucks for North American freight applications should be made from cast materials rather than fabricated components (e.g. side frames, bolster, etc.). Car frames and bodies are to be designed using standard practices as specified by AAR car builders' manuals.

The gondola car will be utilized during severe cold weather conditions. Flame-heated thaw-sheds are employed to aid in the clearing of frozen coal or ore from the cars. Heat-sensitive components and materials should be protected or avoided in new car designs. Interior cross-braces within the tank area should be kept to a minimum and located so as to reduce the possiblity of carried material being caught during the dumping process. Side walls are typically sloped to between 3° and 4°.

Coupling of the car to the locomotive and attachment of all necessary air hoses and control cables should be accomplished using existing train crews. Specialized connectors and equipment, which would require the assistance of an electrician or mechanic to make-up or break the train configuration, should be avoided.

The typical tare or "light" weight of existing gondola cars is about 54,000 lb. Design carrying capacity is approximately 4,100 cu.ft. Loaded or gross weight of cars is generally 263,000 lb, and is regulated by the 36" wheels and axle sizes used by the two 100-ton trucks. Figure 1 is an example of a typical gondola car used by most Canadian railways.

2.2 Hopper Cars

Widely used in unit train services is the 100-ton steel-covered hopper car for the bulk transportation of medium-density commodities (e.g. grain, bauxite, etc.). The cars are of welded construction, typically using high-strength, low-alloy steels. The cars are usually equipped with multiple or full-length trough-type unloading hatches, and four hopper doors are normally used for centre-dumping (see Figure 2).

The external design of the hopper car is less restrictive than that of the gondola car. Its height and length are not limited by existing unloading facilities, but by the car's ability to meet AAR plate "C" clearance requirements.

Typical car tare weight is approximately 62,000 lb, with a nominal carrying capacity of approximately 4,500 cubic ft (grain).

3. POWERED RAILCAR - DESIGN REVIEW

Various design configurations and equipment suppliers were considered in evaluating the feasibility of adapting 100-ton powered rail trucks to the undercarriage of existing gondola and hopper unit train cars.

The availability of two-axle traction-motor-powered trucks with centre plate load-carrying capacities equivalent to standard 100-ton freight trucks is extremely limited. Review of powered truck designs used by transit and freight equipment suppliers indicates that the two-motor, two-axled "GP" or "ZWT" locomotive-style trucks, supplied by General Motors and Bombardier, respectively, are the two major North American standard versions accepted by the railways and adaptable to the axle loads required (see Figure 3).

To allow for adequate top-of-rail clearance and commonality of traction motor equipment used by existing locomotives, 40" diameter wheels and standard 6½" x 12" bearing and axle assemblies would be required on the trucks.

In consideration of the clearance envelope of the powered truck designs, the centre-sill height of the car structure would be required to be raised by approximately 18 inches. Coupler centre height would

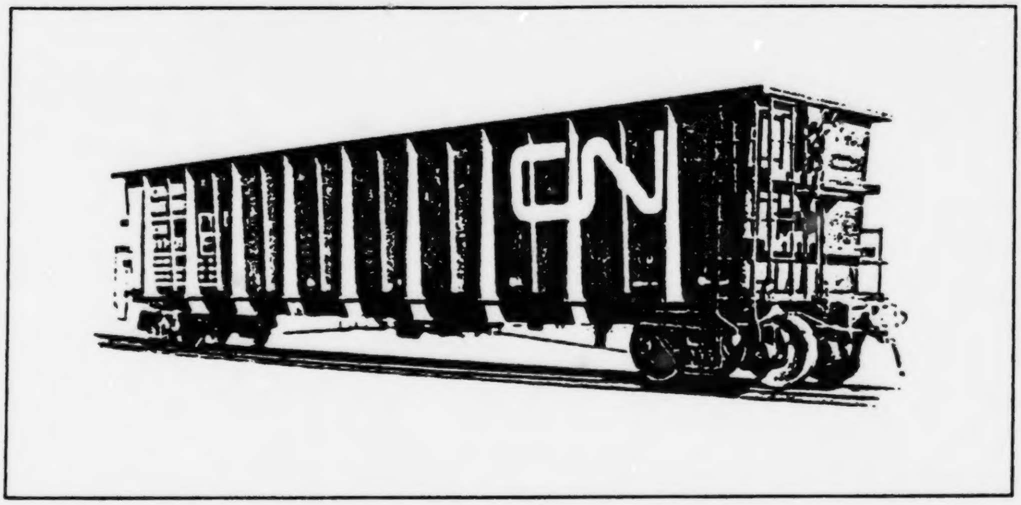


Figure 1: CN Rail 100-ton 4,000 cu.ft. rotary dump gondola car

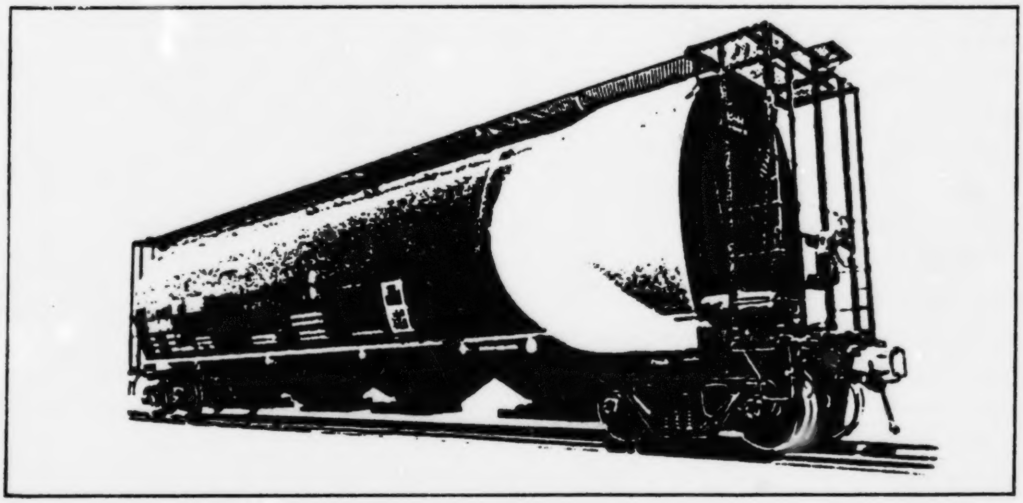
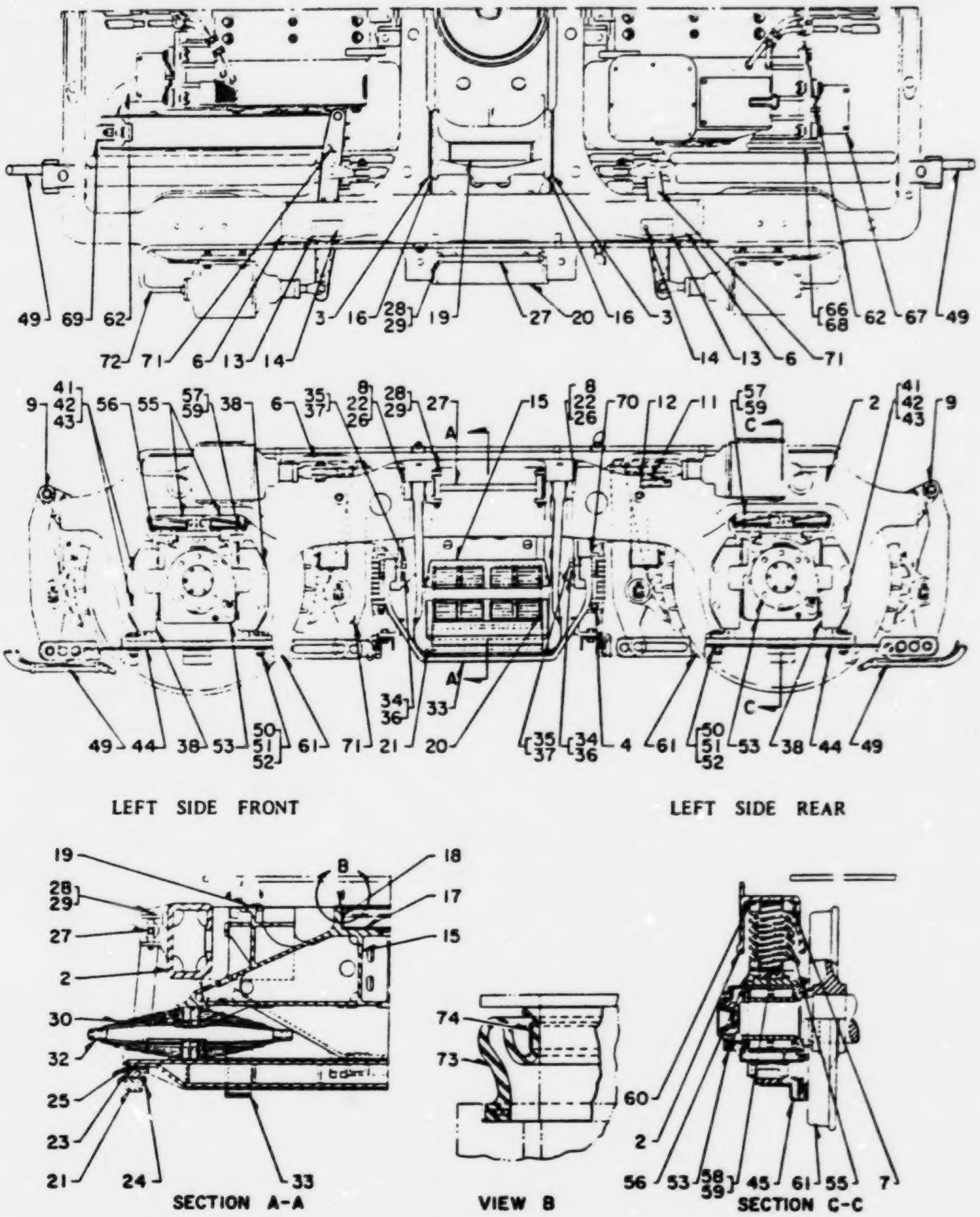


Figure 2: CP Rail 100-ton 4,500 cu.ft. steel-covered hopper car

P.L. No. G1508

TRUCK ASSEMBLY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LEFT SIDE FRONT
LEFT SIDE REAR
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27
2
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32
25
23
21
24
B
18
17
15
33
SECTION A-A
74
73
VIEW B
60
2
56
53
58
59
45
61
55
7
SECTION C-C

Ref. No. 1
PLATE B1508-1

Figure 3 General Motors "GP" locomotive truck (swing-motion with clasp brakes)

be required to be maintained at the standard 34 inches above top-of-rail at both ends of the car, necessitating an offset coupler-to-frame mounting arrangement to be adapted as used on current locomotive designs.

Truck centres, for equivalent coupler centre distances, would be reduced by nearly 8 feet in order to provide sufficient end-of-car clearances.

The additional weight of the GP truck(s) (approximately 14 tons heavier compared to a 100-ton freight truck) will contribute substantially to the vehicle's nominal weight. The total carrying capacity limit of the car would be reduced by the added weight in order to meet most rail load restrictions limits.

Review of the design features outlined, as they pertain to the conversion of the gondola and hopper cars, suggests the following modifications would be required:

- adapt support equipment (blowers) for operation of locomotive trucks
- increase clearance for larger diameter wheels
- raise centre-sill height
- offset coupler mounts
- shorten truck centre distances
- reduce load-carrying capacity.

In addition, when considering the overall height restrictions imposed on existing car designs, it becomes evident that a direct conversion of the cars for powered application is neither practical nor feasible.

Alternatively, redesigned "booster" cars configured to suit the space envelope and application of a bulk commodity freight car appears to be more practical. Two concepts proposed for unit train service which make use of the booster car arrangement are described in the following sections.

3.1 Powered Gondola Car

The practice of employing "booster" cars to supplement train movement is not new. Currently CN Rail and others make wide use of electrically powered slave cars to assist and supplement the tractive effort of yard locomotives during shuttling operations. These units are constructed from a standard locomotive design. Their lower frame and truck arrangement is the same but the engine and car area is removed (see Figure 4). The cars are ballasted with steel and concrete to obtain the necessary loaded weight. In operation, the unit is semi-permanently attached to, and electrified and controlled through, a single locomotive.

The concept proposed for a powered gondola car would be configured for rotary dump service and designed as a "bathtub" style in order to achieve the cubic capacity required (3,100 cu.ft.). The "light" weight of the car, including the weight of the two 38,000 lb GP locomotive trucks, reduces the car's load limit to within 75 percent of conventional gondola cars -- 155,000 lb compared to 210,000 lb. Overall height and coupler centre distances are maintained at present standards (see Figure 5).

A sealed air filtration and control cabinet is located at each end of the vehicle. Located within are two standard locomotive 7 hp, 2,800 cfm, ac-operated blowers used to cool the two traction motors on each truck. Filters in the doors (four per cabinet) are of the 20" x 20", fibreglass, throw-away type commonly used by conventional locomotives. Each traction motor requires about 2,600 cfm of air at 8" H_2O for cooling at its maximum power and duty cycle.

Large-diameter, high-power electrical cables (1.0" diam. for 444 MCM and 1.25" diam. for 500 MCM) would be used to provide the power to the traction motors. Cables would be long enough to permit the dumping operation of the car and would terminate at muliple three-pin high-amperage bulkhead connectors (Pyle-National) attached to an adjacent locomotive. The four D77 traction motors used by the car design are rated for approximately 1,050 amp. each of continuous duty at 1,500 V dc. The number of cables and connectors would be sufficient to permit the duty cycle required, would be environmentally sealed, and would permit acceptable flexibility for train-line hook-up.

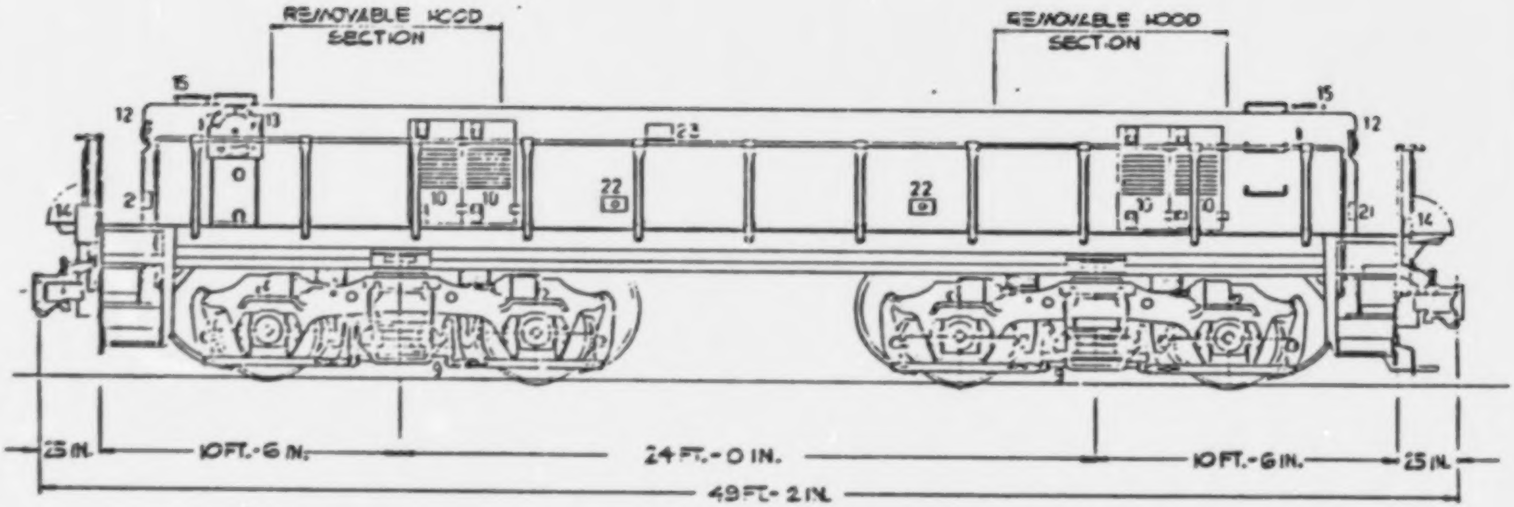
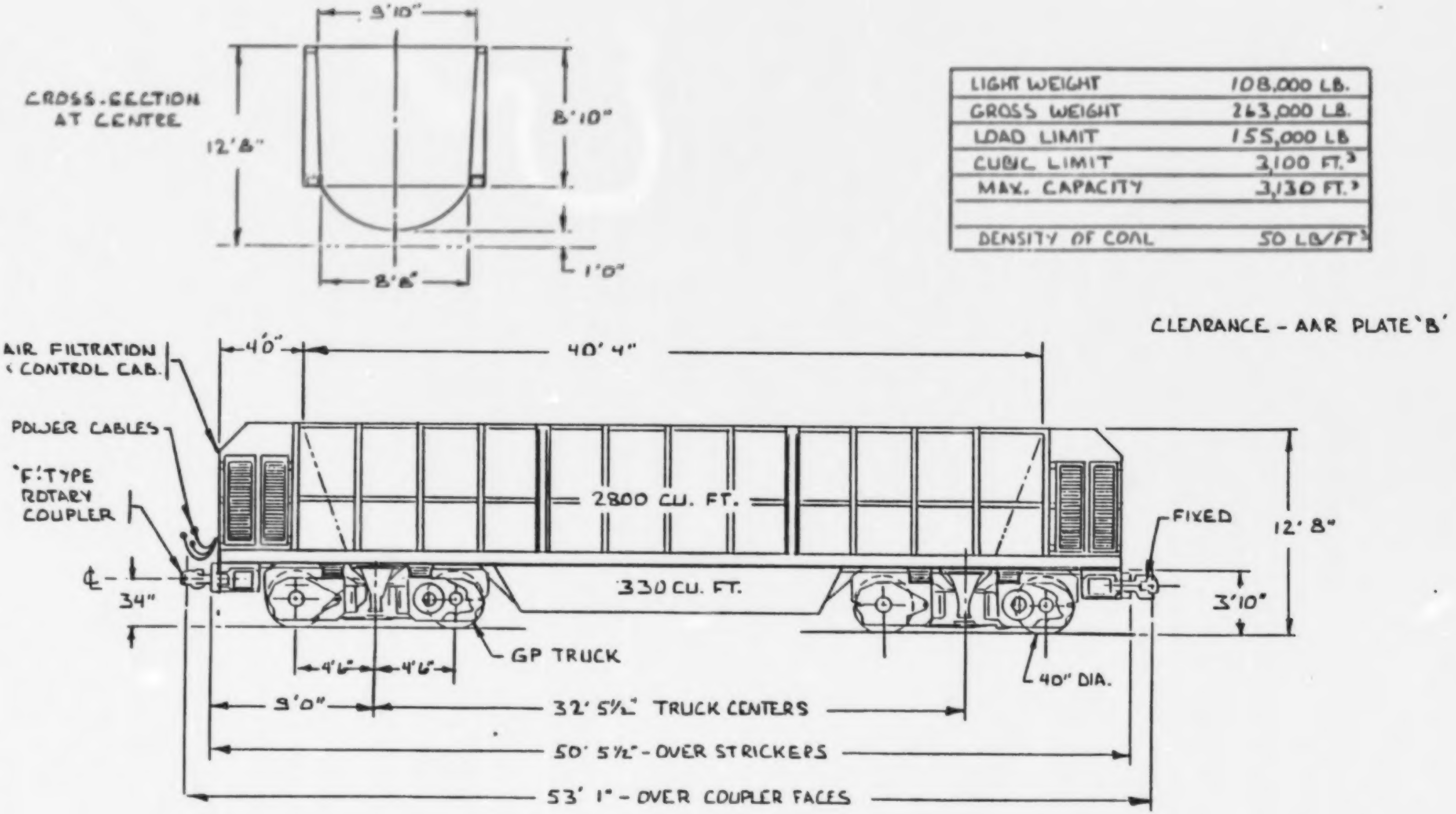

REMOVABLE HOOD SECTION
REMOVABLE HOOD SECTION
15
15
12
12
13
2
21
14
14
10
9
22
22
23
25 IN.
10 FT.-6 IN.
24 FT.-0 IN.
10 FT.-6 IN.
25 IN.
49 FT.-2 IN.

Figure 4 General Motors booster car design


CROSS-SECTION AT CENTRE
9'10"
12'8"
8'10"
8'8"
1'0"
CLEARANCE - AAR PLATE 'B'
AIR FILTRATION & CONTROL CAB.
POWER CABLES
'F' TYPE ROTARY COUPLER
4'0"
40' 4"
2800 CU. FT.
330 CU. FT.
FIXED
12' 8"
3'10"
34"
GP TRUCK
4'6"
4'6"
40" DIA.
9'0"
32' 5½" TRUCK CENTERS
50' 5½" - OVER STRICKERS
53' 1" - OVER COUPLER FACES

LIGHT WEIGHT	108,000 LB.
GROSS WEIGHT	263,000 LB.
LOAD LIMIT	155,000 LB
CUBIC LIMIT	3,100 FT.³
MAX. CAPACITY	3,130 FT.³
DENSITY OF COAL	50 LB/FT³

Figure 5 HPIT 100-ton powered gondola car

General Motors "GP"-type two-axle trucks are suggested for application to this car design. The truck design recommended would be the swing-motion style, exhibiting a leaf-spring secondary suspension and configured with two-shoe clasp brakes. Most major Canadian railways prefer this truck design over competitors' models and their maintenance facilities can more readily accommodate the units (i.e. 90 percent of all locomotive trucks are from GM). As previously noted, 40" diameter locomotive-style wheels and standard 6½" x 12" axle assemblies will be utilized.

3.2 Powered Hopper Car

The adaptation of powered trucks under a redesigned 100-ton steel-covered hopper car would follow similar structural changes and equipment requirements as previously outlined for the gondola version. The concept shown in Figure 6 would have a "light" weight of approximately 59 tons and a maximum cubic capacity of 3,300 cu.ft. (grain). The clearance outlined and vehicle length would follow similar dimensions currently in use. The concept would be configured with three rather than four bottom unloading hatches.

The hopper car would be suitable for bi-directional operation and feature an air-filter and control cabinet at both ends. The unit would be operated and controlled in a married-pair configuration with a suitably adapted unit train locomotive.

4. CONCLUSION AND RECOMMENDATIONS

A study was carried out to evaluate the feasibility of a direct conversion of existing bulk commodity freight cars (i.e. gondola and covered hopper cars) to traction-powered units. Restraints on specified overall vehicle clearances and designs of trucks available resulted in the conclusion that the adaptation of existing cars would not be feasible for further design considerations.

The study does, however, propose the consideration of redesigned freight cars configured similar to existing slave-operated booster cars. The economic benefit of employing these new cars with reduced carrying capacity in a unit train service is recommended for further study.

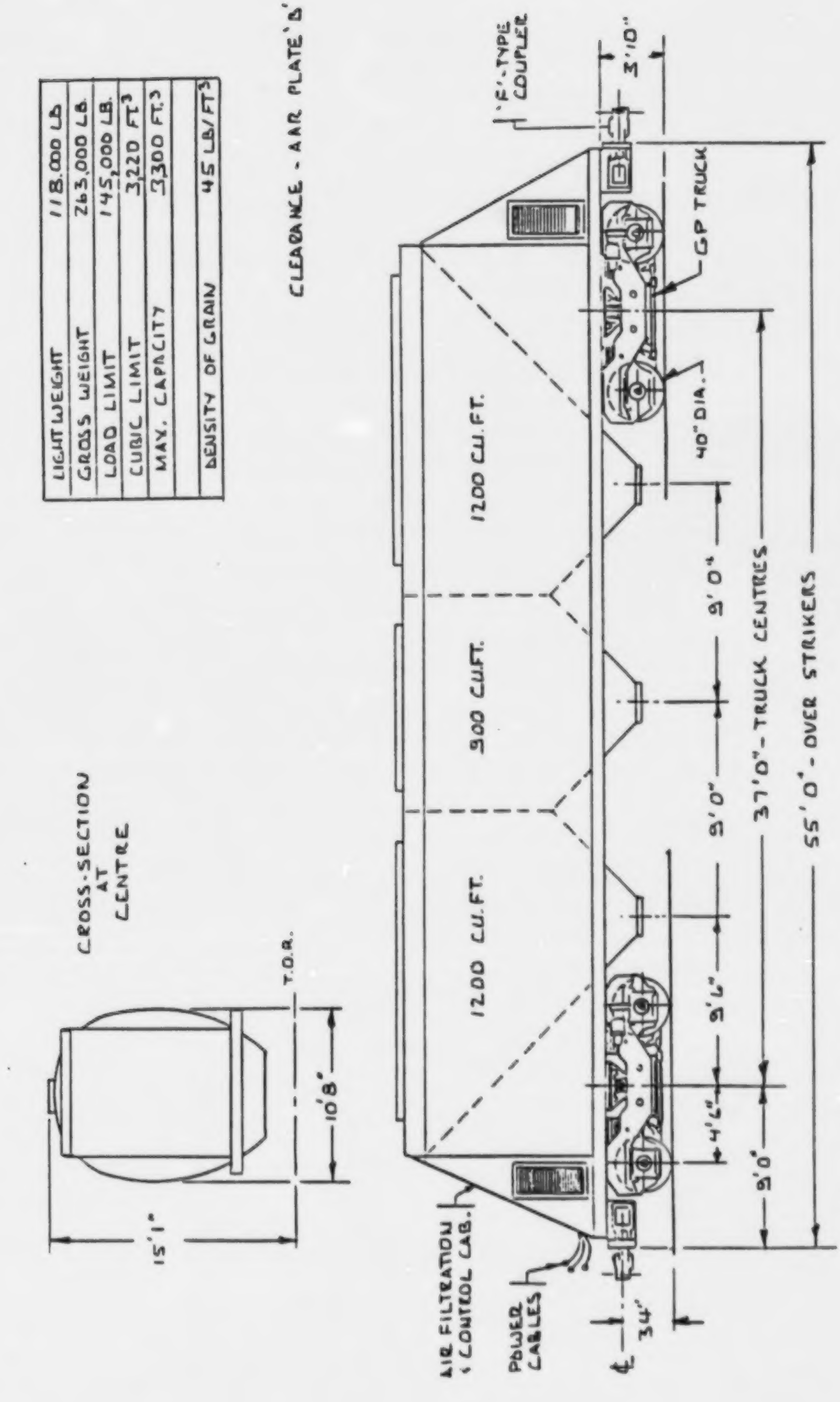

LIGHT WEIGHT 118,000 LB
GROSS WEIGHT 263,000 LB
LOAD LIMIT 145,000 LB
CUBIC LIMIT 3,220 FT³
MAX. CAPACITY 3300 FT³
DENSITY OF GRAIN 45 LB/FT³
CLEARANCE - AAR PLATE 'B'
CROSS-SECTION AT CENTRE
T.O.R.
15'1"
10'8"
'F'-TYPE COUPLER
3'10"
GP TRUCK
40" DIA.
1200 CU.FT.
900 CU.FT.
1200 CU.FT.
9'0"
9'0"
9'6"
4'6"
9'0"
37'0" - TRUCK CENTRES
55'0" - OVER STRIKERS
AIR FILTRATION & CONTROL CAB.
POWER CABLES
34"

Figure 6 HPIT powered steel-covered hopper car

APPENDIX I

MISCELLANEOUS LOCOMOTIVE DESIGN AND PERFORMANCE CALCULATIONS

prepared for

The Canadian Institute of Guided Ground Transport

by

David D. Taylor
David D. Taylor & Associates

Disclaimer

This report has been prepared by the author based on information available to him and on best estimates where exact data were not available. All estimates have been identified as such in the report and the supporting calculations. The results given are sufficiently rigourous to be used with confidence in feasibility of conceptual studies, but should not be used as a basis for detailed engineering design work.

January 26, 1987

I.1. WEIGHT REMOVAL POTENTIAL FOR LOCOMOTIVE FRAMES

The present locomotive desgn was analysed to determine the weight savings possible by reducing the amount of force the center sill is required to withstand. The present and proposed design requirements are given below

(1) Simplified Load Criteria Used by North American Manufacturers

A) Loading

	Load	Amount	Source
(i)	Buff	1,000,000 lb	AAR
(ii)	Draft	450,000 lb	historic appr4Fimately equal to strength of knuckle
(iii)	Lift one end at bumper plate, truck hanging, other end supported at the truck bolster.		

B) Safety Factors

Combination	SF
Weight + buff	1.00
Draft + 1.25 static	1.667
Lift one end + truck	1.25

(2) Proposed Relaxed Design Criteria

A) Loading

	Load	Amount
(i)	Buff	330,000 lb
(ii)	Draft	330,000 lb
(iii)	Lift locomotive only at bolster, truck hanging.	

B) Safety Factors

Combination	SF
Weight + buff	1.00
Draft + Static	1.00
Lift at Bolster	1.00

I.1.2 Analysis of Static Loads

1) Simplified Beam Model

For this analysis it was assumed that all weight except that of the engine, alternator and fuel tank is uniformly distributed. The overall length of the locomotive is 785 inches, the truck center distance is 525 inches, and the fuel tank is centered between the trucks and the supports are 244 inches apart.

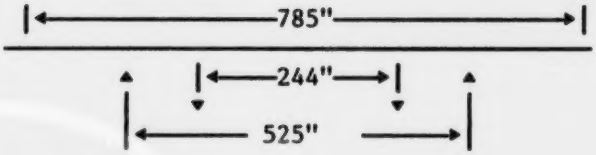


Figure I.1

The weight of the fuel tank is 11,500 lbs empty, and carries 27,800 lbs of fuel. The engine weight is 25000 lbs, and the alternator weight is 35000 lbs at the alternator. The distance between the longitudinal center of gravity of the two components was taken to be 129", and the combination was positioned so that each truck is equally loaded. The engine & alternator position is as calculated below.

(i) Locate engine to balance frame (for this calculation exercise)

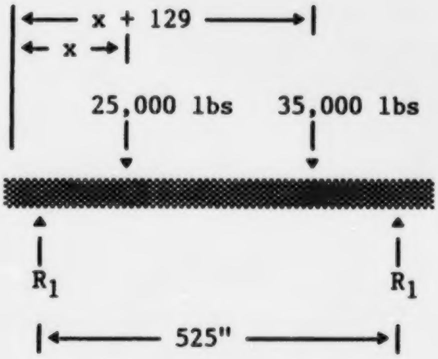


Figure I.2

Σ of vertical forces

$$2R_1 = 60,000 \text{ lb}$$
$$R_1 = 30,000 \text{ lb}$$

Σ of moments about front bolster

$$525\ R_1 - (25{,}000\ x + 35{,}000\ (x + 129)) = 0$$

$$525\ (30{,}000) - 60{,}000\ x - 4{,}515{,}000 = 0$$

$$x = 187''$$

(ii) Distributed load

axle load	390,000
trucks, 2 at 57,500	-115,000
fuel and tank	-39,300
engine and alternator	-60,000
distributed weight	175,700

$$W = \frac{175{,}700 \text{ lbs}}{785 \text{ inches}} = 223.8 \text{ lb/in.}$$

(iii) Static Load on Center Sill

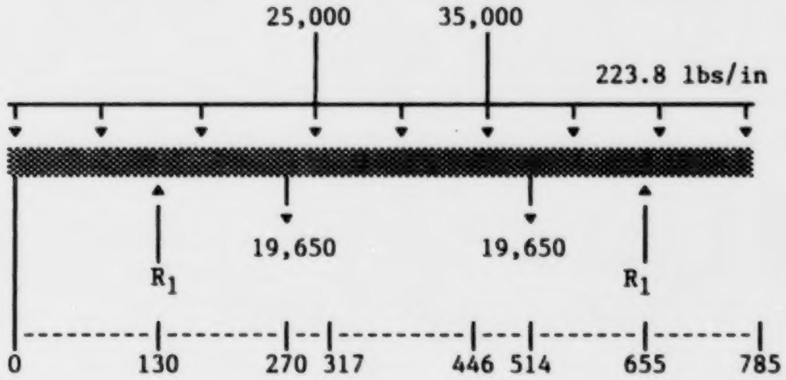

25,000
35,000
223.8 lbs/in
19,650
19,650
R_1
R_1
0
130
270
317
446
514
655
785

Figure A.3

Note : all dimensions are in inches, all forces are in pounds.

R1 = (390,000 - 115,000)/2 = 137,500 lb

Shear and Moments in Beam Due To Static Forces

Given below are the shear and moment in the center sill at various positions along its length. The values for only one-half of the length are given since the loading is approximately symetrical. The stations are the measurement along the length of the sill from the front end of the locomotive, as shown in figure A3.

(iv) Shear

Station	Equation	Shear (lbs)
785	-	0
<655	224 (785-655)	+ 29,120
>655		-108,380
<514	224 (785-514) - 137,500	- 76,796
>514	-76,796 + 19,650	- 57,146
446	224 (785-446) - 137,500 + 19,650	- 41,914
415	224 (785-415) - 137,500 + 19,650 + 35,000	0

(v) Bending moment (sign convention + = top fibres tension

Station	Equation	Moment in lb
785		0
655	(1/2) x W x (785-655)2	1,892,800
600	(1/2) x W x (785-600)^2 - 137,500x(655-514)	-3,729,300
514	(1/2) x W x (785-514)^2 - 137,500x(655-514)	-11,162,108
446	(1/2) x W x (785-446)^2 - 137,500x(655-446) + 19,650 (514 - 446)	-14,530,148
415	(1/2) x W x (785-415)^2 - 137,500x(655-415) +19,650x(514-415) + 35,000x(446-415)	-14,636,850

I.1.3 Analysis of Externally Applied loads

1) Existing Requirements

(i) Moments due to Buff & Draft Forces (approximate)

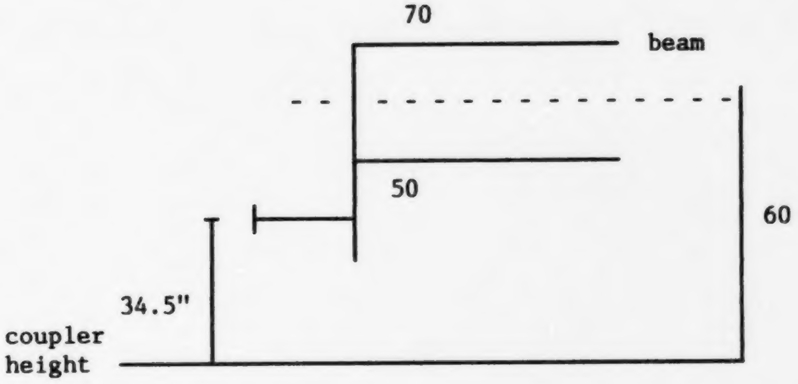
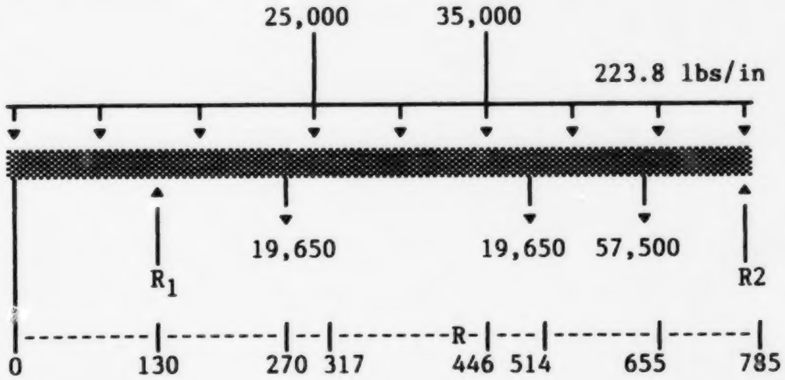

70
beam
50
60
34.5"
coupler
height

Figure I.4

Buff 1,000,000 lbs x (60 - 34.5) inches = 25,500,000 in.lbs

Draft 450,000 lbs x (60 - 34.5) inches = -11,475,000 in.lbs

ii) Forces on Center Sill when Locomotive is Lifted at One End


25,000
35,000
223.8 lbs/in
19,650
19,650
57,500
R_1
R2
R-
0
130
270
317
446
514
655
785

Note : all dimensions are in inches, all forces are in pounds.

Figure I.5

Solving for R1 & R2

Σ of vertical forces

$$R_1 + R_2 = 390{,}000 - 57{,}500 = 332{,}500$$

Σ of moments about front bolster (R1)

$$655\ R_2 + (130)^2 \left(\frac{224}{2}\right) = 57{,}600\ (655 - 130) + 19{,}650\ (514 - 130)$$

$$+\ 35{,}000\ (446 - 130) + 25{,}000\ (317 - 130)$$

$$+\ 19{,}650\ (270 - 130) + (785 - 130)^2 \left(\frac{224}{2}\right)$$

$$R_2 = 156{,}300 \text{ lb}$$

$$R_1 = 176{,}200 \text{ lb}$$

Shear in Center Sill

Station	Equation	Shear
785	R_2	156,300
655	156,300 - 57,500 - 130 (224)	69,680
514	156,300 - 57,500 - 19,650 - (271)(224)	18,446
>446	156,300 - 57,500 - 19,650 - (329)(224)	5,454
<446		-29,546

Bending Moment

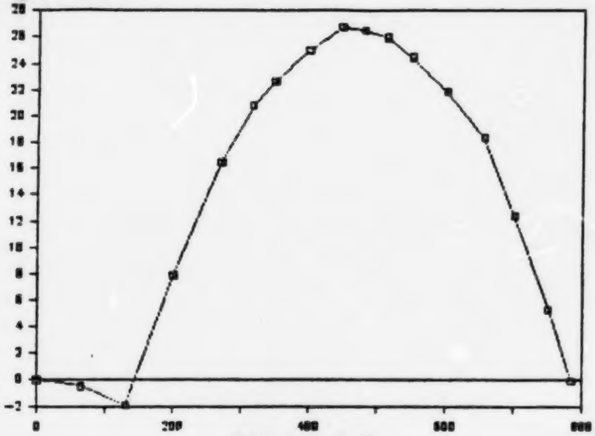


Figure I.6

Station	Equation	Moment
785	-	0
655	R_2 (785-655) - W (785 - 655)^2/2	18,426,200
600	R_2 (185) - W (185)^2/2 - 57,500 (55)	21,919,800
514	R_2 (271) - W (271)^2/2 - 57,500 (141)	26,024,408
446	R_2 (339) - W (339)^2/2 - 57,500 (209) - 19,650 (68)	26,760,848

iii) Summary of Moments on Center Sill

Station	Static in.-lb x 10^6	Buff in.-lb x 10^6	Draft in.-lb x 10^6	End Lift in.-lb x 10^6
785	0.00	25.5	-11.5	0.0
655	+1.89	25.5	-11.5	18.4
600	-3.73	25.5	-11.5	21.9
514	-11.16	25.5	-11.5	26.2
446	-14.53	25.5	-11.5	26.8
415	-14.64	25.5	-11.5	26.0

iv) Combined Loads on Center Sill

The combined loading cases to be considered are as follows:

a) Static + Buff with a factor of safety = 1.0
b) Draft + 1.25 x static with a factor of safety = 1.66
c) Lift one end with a factor of safety of 1.25

The values in the table below are adjusted to account for the required safety factors. The assumed yield strength of the metal is 50,000 lbs/in.2

R

Station	Static +Buff in.-lb x 10^6	1.25xStatic +draft in.-lb x 10^6	End Lift in.-lb x 10^6	Worst Case in.-lb x 10^6	Moment of Area Req'd in.3
785	25.5	-18.4	0.0	25.5	510
655	27.4	-15.2	23.0	27.4	548
600	21.8	-27.4	27.4	27.4	538
514	14.3	-42.4	32.5	42.4	848
446	10.9	-49.4	33.4	49.4	988
415	10.9	-49.6	-11.5	49.6	993

(v) Conventional Combined Beam Requirement

The moment of area of the beam required is as shown in the following figure:

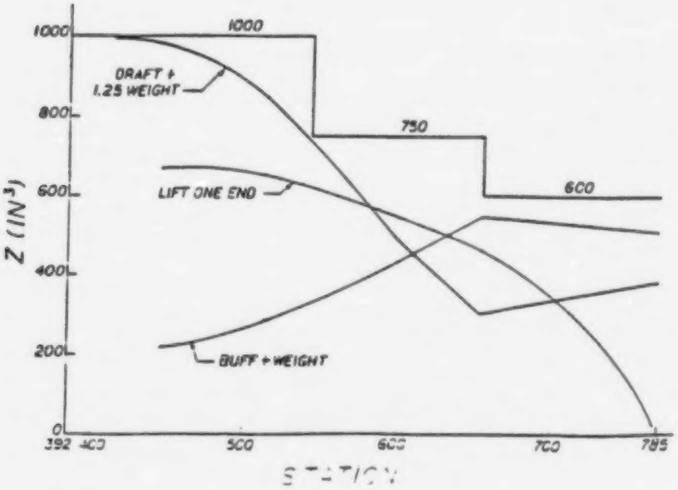


Figure I.7

Arbitrary units of Z x for each section:

= 1000 (500-392) + 750 (105) + 600 (130)

R

= approximately 315,000 units.

2) Reduced requirement Center Sill

The requirements for the reduced weight center sill are as follows:

Bending Moments:

weight:	unchanged
buff:	= 330,000 (60 - 34.5) = 8,415,000
draft:	= -8,415,000

The shear and moment in the center sill due to static loads are the same as for the conventional requirements. The loads in the center sill for lifting of the locomotive would be the same as the static load, since the force required to lift the hanging truck would only effect the bolster load. The moments in the sill due to the various loads are as listed in the following table.

Station	Static in.-lb $\times 10^6$	Buff in.-lb $\times 10^6$	Draft in.-lb $\times 10^6$
785	0.00	8.4	-8.4
655	+1.89	8.4	-8.4
600	-3.73	8.4	-8.4
514	-11.16	8.4	-8.4
446	-14.53	8.4	-8.4
415	-14.64	8.4	-8.4

iv) Combined Loads on Center Sill

The combined loading cases to be considered are as follows:

a) Static + Buff with a factor of safety = 1.0
b) Static + Draft with a factor of safety = 1.0

The values in the table below are adjusted to account for the required safety factors. The assumed yield strength of the metal is 50,000 lbs/in.2

Station	Static +Buff in.-lb x 10^6	Static +draft in.-lb x 10^6	Worst Case in.-lb x 10^6	Moment of Area Req'd in.3
785	8.4	- 8.4	8.4	170
655	10.3	- 6.5	10.3	207
600	4.7	-12.1	12.1	242
514	-2.7	-19.5	19.5	391
446	-6.0	-22.9	22.9	459
415	-6.1	-23.1	23.1	461

The moment of area required for the new requirement center sill is as shown below:

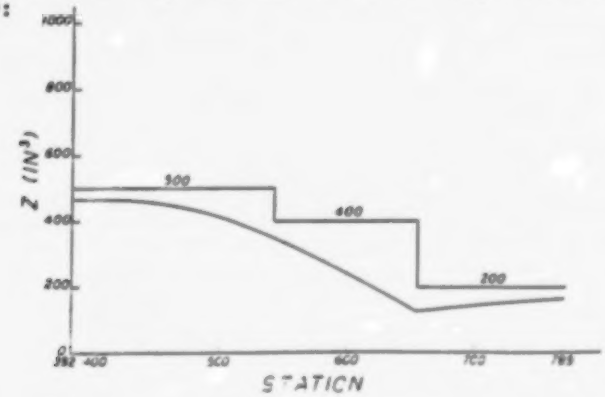


Figure I.8

Arbitrary units = 500 (525 - 392) + 400 (130) + 210 (130)

= 145,800 Z-inch units

(xi) Summary

$$\text{Ratio of Z-inch totals} = \frac{145{,}800}{315{,}000} = 46 \text{ percent}$$

For same beam height, the weight reduction of the beam is about 80 percent of the Z section reduction, because of the constant flange weight:

∴ 46 percent/0.8 = 67 percent

∴ approximate weight reduction - 43 percent

For beam assumed in this study, with full weight reduction but no relaxation of criteria (based on another study):

(1) total frame weight: 53,000 lb

(2) actual frame weight: -30,000 lb

(3) non-beam structure 23,000 lb

Saving on beam: 30,000 x .43 = 12,900 lb

Estimate on non-beam: 23,000 x .30 = 6,900 lb

19,800 lb

(assumes that draw pockets, bumper plates, bolsters, cover plates, etc., can all be reduced for a lighter load application)

New beam weight = $\boxed{31,000}$

Caution: This is an export beam weight with more than the export weight on the frame. Deflections could be too high.

(xii) Beam Deflection

(1) Conventional beam

To calculate I, assume 22" beam where Z = 1,000 and 800

20" beam where Z = 600

$$I = \frac{Z \cdot \text{depth}}{Z}$$

I 1,000 = 11,000 x 1.25* = 13,750

I 750 = x 1.25 = 10,300

I 600 = 6,000 x 1.25 = 7,500

Area	Length	Moment EI (2)	Factor	Area (1)(2)	Moment Amount	Deflection
1	150	27 x 10-6	1	0.0040"	1,875"	0.750"
2	150	7 x 10-6	2/3	0.00070"	212"	0.1490"
3	112.5	35.6 x 10-6	1/2	0.0020"	75'	0.150"
4	130	8 x 10-6	1/3	0.00037"	433"	-0.016"
				Deflection		1.033"

(ii) Beam with Reduced Design Criteria

Bending moments the same, I values change from page 9.

I500 = Z500 x 11 x 1.25 = 6,875

I400 = Z400 x 11 x 1.25 = 5,500

I200 = Z200 x 10 x 1.25 = 2,500

With reference to page 11, deflections vary with I value.

Area	Conventional Beam Deflection	I Ratios	
1	0.750	13,750/6,875	1.500
2	0.149	10,300/5,500	0.279
3	0.150	7,500/2,500	0.450
4	0.016	7,500/2,500	-0.048
			2.181"

---

* 1.25 makes allowance for stiffening effects of the top and bottom plates

Note: The deflections are summarized as follows:

	Static Deflection
Conventional beam (existing):	≈ 0.600"
Lightweight beam - AAR standards:	≈ 1.03
Lightweight beam - UIC standards: (North American Weights)	≈ 2.18

In my opinion, 1" is probably a practical upper limit for deflection; the lightweight beam would be too flexible.